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**University Noise Research - Proceedings of the
EPA-University Noise Seminar Held at Purdue University,
West Lafayette, Indiana on October 18-20, 1976**

Purdue Univ, Lafayette, Ind. Ray W. Herrick Labs.

Prepared for

**Environmental Protection Agency, Washington, D.C. Office of Noise
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UNIVERSITY NOISE RESEARCH

PROCEEDINGS
OF THE
EPA—UNIVERSITY NOISE SEMINAR
OCTOBER 18-20, 1976

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16. ABSTRACT Material presented and discussed at the seminar included research, development and demonstration projects in all areas of noise control except noise generated by aircraft. Emphasis was placed on non-Federally funded programs and hardware oriented projects. An appendix to the report provides an extensive compendium of university noise research projects regardless of the nature of the source or sponsoring institution.		
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PROCEEDINGS
OF THE
EPA-UNIVERSITY NOISE SEMINAR

OCTOBER 18-20, 1976

PURDUE UNIVERSITY
WEST LAFAYETTE, INDIANA 47907

Organized by.....

Ray W. Herrick Laboratories,
School of Mechanical Engineering, Purdue University

Sponsored by.....

Environmental Protection Agency,
Office of Noise Abatement and Control

General Chairman: Raymond Cohen, Purdue University
Program Chairman: David R. Tree, Purdue University
EPA Coordinator: Harvey J. Nozick, Environmental Protection Agency
Co-Editors: Joseph W. Sullivan & Andrew F. Seybert, Purdue University

FOREWORD

In this first (hopefully there will be more) joint EPA - University Noise Research Seminar, it became clear to the organizers of the seminar that all the time and effort expended was well worth it. To see government and university groups in frank, open discussion on mutual problems was indeed gratifying. Whether the larger aims of the seminar will be achieved remains to be seen.

The purpose of the seminar, initiated and sponsored by EPA, was to help EPA and other government agencies become aware of university noise control sources. It is hoped that an understanding of the nature, scope and results of such projects will be useful to government agencies in both their current programs and in their future planning in implementing the requirements of the Noise Control Act of 1972.

Principal investigators of active, industrially-related research programs at universities were invited to present information about work recently completed (during 1975 or 1976), or in progress, or planned for 1977. Papers were solicited on research, development and demonstration projects in all areas of noise control except aircraft noise. Emphasis was placed on industrially sponsored, hardware oriented projects.

In all, 39 people attended: 29 from universities, 8 from government and 2 with other affiliations. A total of 23 papers were presented. It is regrettable that a few professors working in this area were unable to respond to the call for papers.

In these proceedings, it was felt important to capture the mood and atmosphere of the various sessions. For this reason all discussion was tape recorded. The Editors have endeavored to put as much as possible of this discussion in its original form into the proceedings.

In order to make this volume as useful as possible, we felt it important to provide a list of all university noise control related projects. Thus we invited submission of information about such projects over the last three years. A compilation of material from those who responded is provided in an appendix.

Sincere thanks go to the many people who have made this seminar possible in such a short span of time. Particular thanks go to the authors who responded on time with their papers; to Harvey Nozick and his associates for their coordination efforts at EPA; and to the many who responded with the data that made it possible to assemble the appendix of current university research in noise control.

Raymond Cohen
Andrew F. Seybert
Joseph W. Sullivan
David R. Tree

Purdue University
December, 1976

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Raymond Cohen

Director of Ray W. Herrick Laboratories
Purdue University, West Lafayette, Indiana

I would like to welcome you on behalf of Purdue University, the Schools of Engineering, and all of my colleagues. We are delighted and honored to host this prestigious, elite group. We have virtually all of the universities represented that are doing research in noise control. We omitted the aircraft field because it has been covered in similar conferences and seminars.

Tomorrow afternoon is an unusual session for which I ask you to be prepared. Tomorrow afternoon is set aside for a free-for-all discussion, and we hope that you will be prepared to air your views concerning noise control research, EPA's future regulatory activities, and other related matters.

The title of this seminar is: "The Environmental Protection Agency - University Noise Research Seminar". A general increase in the noise level produced by increased use of noise-producing equipment, and a better understanding of the effects of noise on man have made people realize that one of the major problems facing the United States is reducing the noise level of our environment. The progress that EPA has made in combating environmental noise since enactment of the 1972 Noise Control

Act is certainly significant. This seminar will enlighten the EPA and other government agencies concerning university noise control projects and will provide an understanding of the nature, scope, and results of such projects. This will be beneficial to government agencies in carrying out current programs and in future planning to implement the requirements of the Noise Control Act.

Principal investigators in active research programs at universities were invited to present information concerning their work in noise control--work recently completed, in progress, or planned for next year. Papers were invited on research, development, and demonstration projects in all areas of noise control except aircraft noise. A number of the leading noise control experts at universities in the United States were invited to join about 10 to 15 government representatives to discuss their work and future needs. Emphasis was placed on hardware oriented noise control projects, and programs sponsored by industry received priority. Government-sponsored projects will be listed, and some remarks made about them, in the compendium that will constitute the appendix of the proceedings.

John Schettino, from the Office of Noise Abatement and Control will be your next speaker.

EPA OBJECTIVES

John Schettino

Director of Noise Control Requirements in Technology Staff,
Office of Noise Abatement & Control, Environmental Protection Agency

Thank you, Ray. I want to welcome you on behalf of the Environmental Protection Agency. This seminar is a very important element of our plan to present to the Congress a comprehensive report on all ongoing noise research, not only research

supported by the Federal Government but also research supported by universities under private grants, research conducted by industry using their own resources and financing, and international research applicable to the needs of the Federal Government.

I think it might be appropriate to give a bit of background on what has led us to support a seminar by Purdue to try to obtain the information on ongoing non-Federally funded research at universities. I guess my title is a good place to start. I'm Director of Technology Requirements and Federal Programs Division of EPA's Office of Noise Abatement and Control. One might wonder how technology requirements and Federal programs come together. The answer lies in the construction of the Noise Control Act of 1972, in that the Congress did not envision that the Environmental Protection Agency would compete with the sizeable capability already developed within the Federal establishment. That capability is represented by the Department of Transportation, the National Aeronautics and Space Administration, and some 25 other Federal agencies that had responsibility for noise abatement and control prior to the enactment of the Noise Control Act of 1972. Coordination of this effort was desirable and necessary. As a matter of fact, had the responsibilities of those other 25 agencies been properly discharged, there would have been no need for a Noise Control Act or an Office of Noise Abatement and Control.

My job is to establish the technology requirements for the Environmental Protection Agency. I accomplish this by coordinating the activities of other Federal agencies and encouraging them to budget as necessary to satisfy our requirements, and to undertake the R & D necessary to support our regulatory activities. Early in 1974, the Environmental Protection Agency initiated its Federal coordination responsibility. It gives me a lot of pleasure to acknowledge the presence of Dr. Frank Hart, from North Carolina State University, who was instrumental in establishing that program in the early days of EPA when he was employed by the Office of Research and Development of EPA, who had the responsibility at that time for coordination of the Federal R & D. As a result of approximately 18 months of effort on the part of the Office of Research and Development, it was possible for the Environmental Protection Agency to publish and submit in mid-1975 to the Congress a report on all ongoing Federal research dealing with noise.

Time did not permit a comprehensive evaluation of that ongoing work to establish its relevance to the needs of EPA and to form the basis for additional requests of the Congress for funding to support the needed R & D. We are now in the process of doing that, and we thought it advisable to up-date the inventory developed in the 1973-1974 period. We also felt it advisable to include other research that is not covered by Federal funds.

In 12 to 18 months we hope to publish a number of reports summarizing all of the ongoing noise research. More important than just issuing an inventory, there would be analysis of the relevancy of that noise research to EPA and other Federal establishment requirements. For the first time, all Federal agencies will have a better understanding of what other agencies are doing relevant to their needs. Right now, there is no mechanism to bring together this body of data and the Federal experts who are sponsoring this research. The only program, of which we are aware, that attempted to coordinate Federal research was undertaken in aviation. That body was disbanded several years ago and was not replaced until the Federal coordination program was initiated by EPA's Office of Research and Development. Now, we hope to carry out more aggressively the mandates of the Noise Control Act. I'm pleased we have the resources to do that, and those resources have made it possible to support this seminar.

I'd like to close my remarks by announcing that sometime within the next 30 days the Environmental Protection Agency will make available, in the Federal Register, a strategy for a National Noise Abatement and Control Program. I want to repeat that it is a National--not an EPA--program. It will cover all aspects of the noise problem, not only pieces for which EPA has responsibility. We encourage the public, industry, and universities to submit their comments, recommendations, and critique on this strategy document. It is the desire of the Administrator of EPA to permit the widest possible participation in EPA's business of establishing programs to abate and control environmental pollution, including noise. I encourage this body to submit its comments after reviewing the document.

LAWN TRACTOR NOISE REDUCTION

David R. Tree
Purdue University
West Lafayette, Indiana 47907

INTRODUCTION

In a report by M.J. Rudd and E.K. Bender [1] prepared for the U.S. Environmental Protection Agency, the major noise sources of lawn mowers and their levels were reported. This study gives noise level information about three types of lawn mowers: a) reel mower 2 to 2-1/2 hp, b) the walk-behind rotary mower 3 to 5 hp, c) the riding rotary mower 5 to 8 hp. The report completely neglected the larger and ever increasing so called lawn and garden tractor with mowing attachments.

Two interesting findings other than levels of rotary mowers are reported in reference 1 the first which agrees with our findings and the second which does not: 1) the noise level of the reel mower is much less than the rotary mower, and at this stage there is no need to try to reduce the noise level of the reel mower and 2) the sharpness of the rotary blade has little to do with the cutting process.

Figure 1 reproduced from reference 1 shows the major noise sources reported for a gasoline-powered rotary mower. Note that for both types the major noise source is the exhaust noise.

For several years now the Ray W. Herrick Laboratories has been studying the noise level produced and reduction methods for lawn and garden tractors 5-18 hp with rotary mowing attachment. At the same time the Agricultural Engineering Department of Purdue University [2,3] has been studying the effect of blade sharpness, shape, speed, etc. on the cutting quality and ability. Because of the confidential nature of this work, it is not possible to give much detail about the cutting quality work. The author does know that they have found that blade sharpness, speed, and design all can have a strong effect on cutting efficiency and quality. Applegate and Crocker [4] have also shown that these factors can have an effect on the noise produced by the mower.

DESCRIPTION OF TRACTOR AND TESTS

Noise source identification programs were conducted on two single cylinder, sixteen horsepower lawn and garden tractors. Both were air cooled, four cycle engines. Tractor A had a hydrostatic transmission and side shields were used as partial acoustical enclosure. Tractor B had a six speed geared transmission and no side shields. Both tractors could be fitted with a three bladed rotary mower.

Measurements

In order to better understand the data presented, a brief description of the data taking processes is needed. All measurements reported were taken in a reverberation room. The tractors were mounted on a dynamometer capable of loading the tractor through the wheels and transmission or directly through the engine.

In trying to determine the noise sources of each tractor, the noise level of the following components were examined: 1) engine, 2) exhaust, 3) air intake, 4) transmission, 5) mower deck, and 6) where possible vibration of metal surfaces.

In reporting the data in this paper, the so called engine noise and transmission noise also includes some noise for metal parts which were caused to vibrate because of driving forces coming from the engine and transmission.

As much as possible the noise source identification was accomplished by testing each component separately or reducing to an insignificant value all noise levels but one.

For example:

The noise level of the mower deck was obtained by driving it with quiet electric motors, exhaust and air intake noise was ducted out of the room, where possible metal parts were removed.

The biggest problem was to reduce the engine noise level to low enough values so that other components could be measured. For one tractor, the engine was water cooled by placing copper tubes inside the air fins and then the engine was wrapped with a lead sheet.

In many cases it was still necessary to use the logarithmic reduction method to obtain the noise level.

For example in order to obtain the exhaust noise, all noise sources were eliminated except the engine. It's noise level was measured. The exhaust noise and engine noise were then recorded. The exhaust noise was then determined by logarithmically subtracting the engine noise from the combined noise.

This method has one serious draw back if exact values are needed. If the combined level is not at least 1 dB higher than the engine noise it is impossible to determine the exact value of the exhaust noise. If one only wants to determine if a noise source is a major source, the method offers no ready disadvantages. Thus, in some of the absolute data presented later, there are frequency ranges where no data can be presented.

DATA AND RESULTS

Effects of Load and Speed

Figure 2 shows a typical plot of the total sound power output of the tractors as a function of speed. A comparison of the absolute value of the two tractors should not be made since they were running under different conditions. The graph does clearly indicate that the noise level produced by both tractors is a strong function of speed.

A comparison of the sound power output as a function of engine load (no load to stalled condition) for several different speeds for both tractors showed that the noise output was a very weak function of load. In most cases where the noise did increase as the load was increased, it could be shown that most of the increased noise was due to increase in the noise produced by some metal part vibrating.

Engine, Exhaust, and Transmission Noise

Figure 3 and 4 show the noise levels of the engine, exhaust and transmission of tractors A and B respectively. The two graphs show that in the higher frequencies, the engine of both tractors is a major contributor of noise.

Both graphs also show that for low frequencies the exhaust noise is important but

at higher frequencies it becomes less important. For tractor A on an A-weighted basis the exhaust noise is 6 dB less than the engine noise. While for tractor B the exhaust noise and engine noise are about the same.

In order to investigate the exhaust and other noise in more detail, one engine was wrapped with lead sheets (as discussed earlier) to reduce its noise level well below that of the unwrapped engine. Figure 5 shows the noise level of the engine before and after wrapping with lead sheets. Because of the problem of the logarithmic reduction method some data for the unwrapped engine is missing.

Figure 6 shows the effect of the muffler for tractor B. This data were taken with the lead wrapped engine, air intake exhausted out of the room, all metal parts possible were removed, and the transmission was disconnected. The first graph shows the noise level without a muffler, the second with the production muffler and the third for the exhaust ducted out of the room.

The graph shows that the muffler does a very good job in the high frequency range. In fact above 1000 Hz it reduced the noise level to a value very close to the wrapped engine noise, almost 10 dB below the present engine. The graph also shows that the muffler does very little at low frequencies. On the A-weighted scale the muffler reduces the noise by about 6-7 dB(A).

The big difference between these two tractors is the transmission used in each. Figure 3 shows that for tractor A, the transmission is the major noise source in the high frequency range and on an A-weighted basis is only 1 dB lower than the engine noise. This plot is for the maximum noise level output of the tractor without mowers. While Figure 4 shows that the transmission noise for tractor B at all frequencies is well below the noise of the engine. Later tests showed that this was not the noisiest transmission condition for this tractor. Although the noisiest condition was only 2 dB(A) higher than the one reported and never more than 3 dB in any 1/3 octave band.

Mower Deck

Figures 7 and 8 show the noise level in 1/3 octaves of both mower decks as a function of speed. Both mowers show about the same levels. Applegate and Crocker [4] have discussed mower noise in detail. It will not be repeated here.

Other Noise Sources

The level of all other noise sources on the tractor are well below those discussed

above and are of little importance at this time. Briefly discussed below are the results of some minor noise sources.

The removal of the side panels from tractor A show only a slight overall increase in dB(A) level. Some 1/3 octave band actually increased because of the vibration of the panels.

While testing the lead wrapped engine of tractor B, all possible metal parts were removed, finders, hood, etc. The removing of these parts produced a reduction in the 200, 250, 315 Hz 1/3 octave band. A 9 dB reduction was noted in the 315 Hz band. There was no effect in the A-weighted value.

For both tractors the air intake noise was very low, even using the lead wrapped engine it could only be measured at frequencies below 1000 Hz.

CONCLUSIONS

The results of this study showed that the noise levels in order of importance for both tractors were

Tractor A

1. engine
2. transmission
3. mower deck
4. exhaust
5. air intake
6. panel vibration of removable parts

Tractor B

1. engine
2. exhaust
3. mower deck
4. panel vibration of removable parts
5. transmission
6. air intake

Some experimentation with the transmission of tractor A indicated that that vibration problem of parts of the tractor may be more important than the two lists indicate. Some experimental data were taken on the transmission outside of the tractor. This data showed that the noise produced by the transmission outside the tractor was much less than the noise of the transmission as mounted in the tractor. This leads to the conclusion that the transmission is driving other parts of the tractor causing them to vibrate and produce noise.

Some measurements of the engine outside the tractor indicates that the tractor body may be acting as a partial acoustical enclosure for the engine.

It is very clear from this work, that the major noise source of both tractors is the engine. It is also clear that until this noise source is reduced, reduction of other noise sources will produce very little noise reduction on an A-weighted level.

Clearly there is a need for lawn tractors to have noise reduction research done on the engine. The author feels that with little work the noise level floor of all types of lawn mowers will be the engine.

It is surprising to this author the small amount of noise reduction work being done on these small engines. Their use is not limited to lawn tractors.

The research work should take two directions. A careful examination of the noise sources of the engine should be undertaken. Once this has been accomplished, recommendation for engine redesign to reduce noise can be given.

A second approach should be the study of enclosures or partial enclosures to reduce engine noise. Our research showed although tractor A has side panels which served as a partial enclosure, they did little to reduce the overall noise level output. There is no question that they did change the directivity pattern of the tractor.

The Herrick Laboratories is working in both these areas and hope to have results to report shortly.

APPRECIATION

The author would like to express appreciation to the following graduate research assistants. All of whom have worked with the author over the years to produce the data reported in this paper. Listed in the references are publications which they wrote from which the data of this paper was taken. They are: David P. Egolf [5], Kathleen M. Mulej [6], Lorne W. Tweed [7,8], and Deane B. Jaeger [9].

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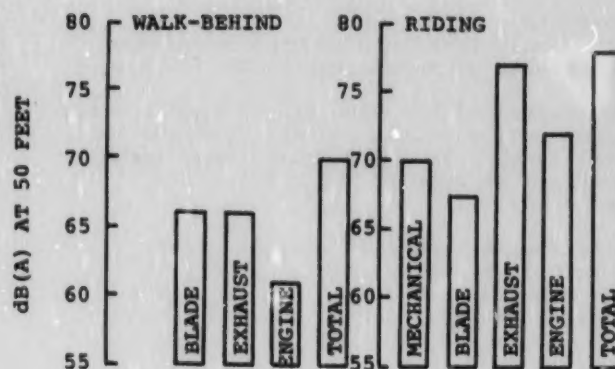


FIG. 1 SOUND SOURCES AND LEVELS OF WALK-BEHIND AND RIDING LAWN MOWERS FROM REF. 1

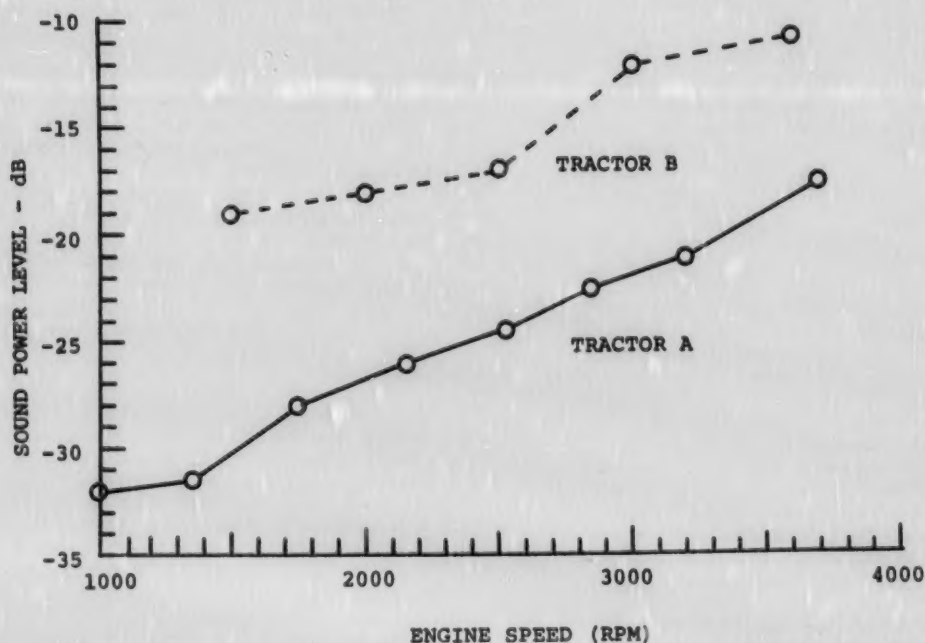


FIG. 2 A-WEIGHTED SOUND POWER LEVEL AS A FUNCTION OF SPEED

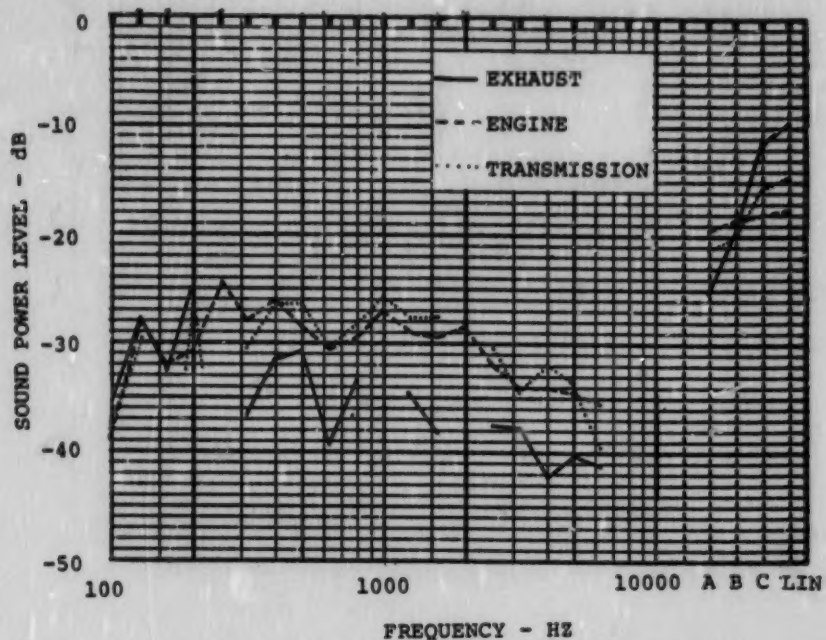


FIG. 3 SOUND POWER LEVEL OF THREE MAJOR NOISE SOURCES
FOR TRACTOR A

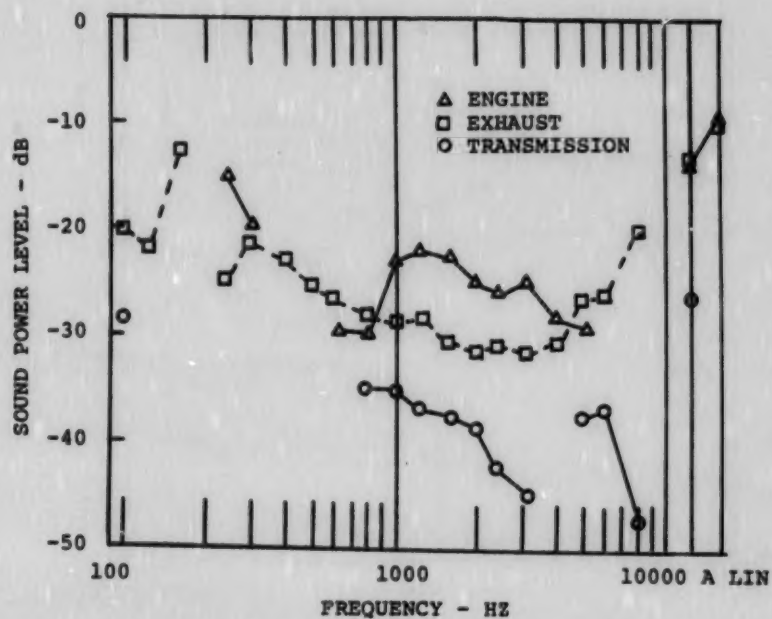


FIG. 4 SOUND POWER LEVEL FOR THREE SOURCES FOR TRACTOR B

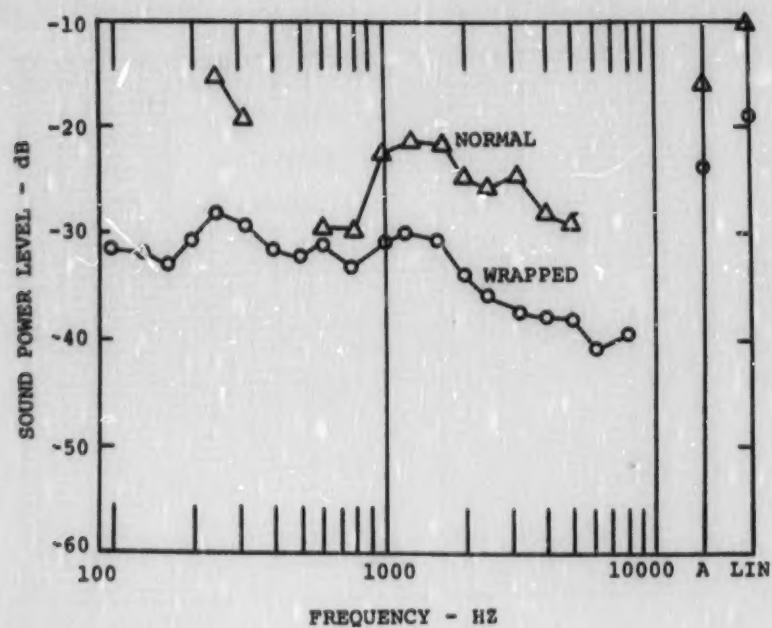


FIG. 5 COMPARISON OF ENGINE NOISE OF TRACTOR B
BEFORE AND AFTER WRAPPING WITH LEAD SHEETS

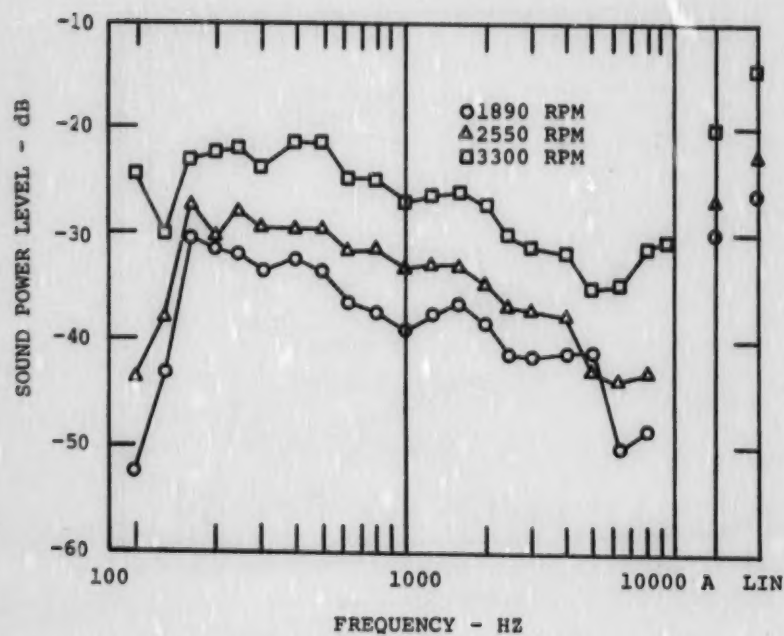


FIG. 6 MOWER DECK NOISE VERSUS SPEED - TRACTOR B

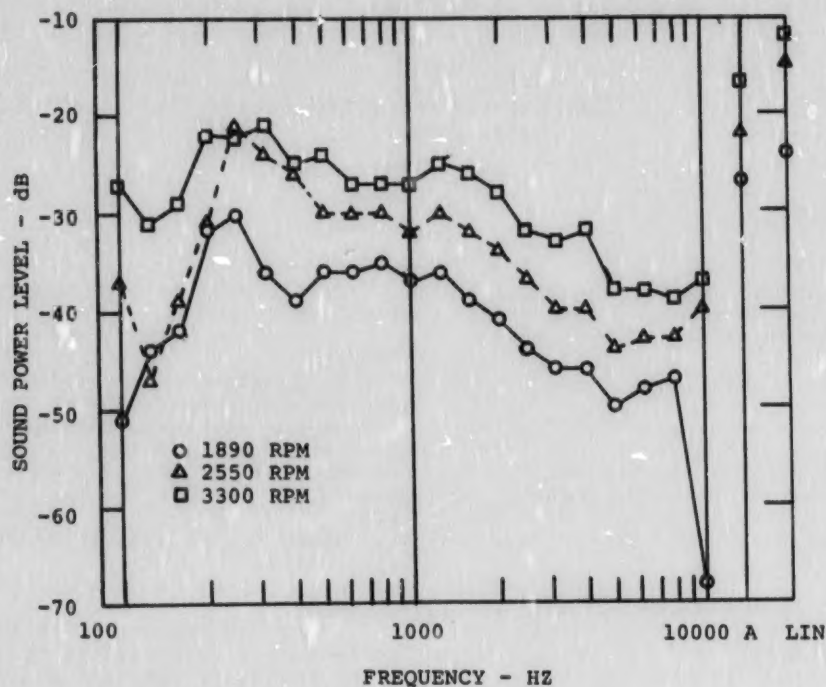


FIG. 7 MOWER DECK NOISE VERSUS SPEED - TRACTOR A

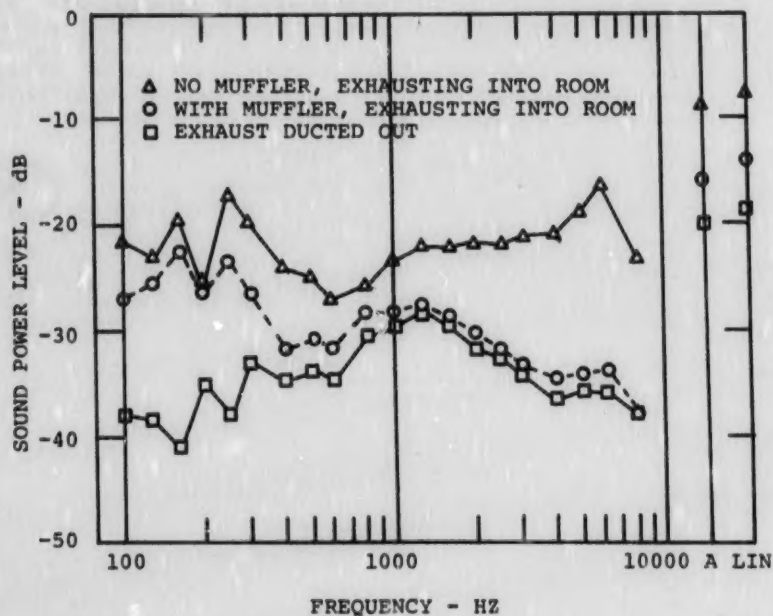


FIG. 8 EFFECT OF MUFFLER ON WRAPPED ENGINE - TRACTOR B

"Lawn Tractor Noise Reduction"
presented by

D.R. Tree

DISCUSSION:

D. Muster:

The first slide you showed had two lawn mower types you were comparing. What were the similarities or dissimilarities between the two?

D.R. Tree:

The first slide was reproduced from an EPA report which was prepared by Bolt Beranek & Newman. What they did was take a large number of walk-behind and riding mowers, which were less than about 8-12 HP, and the results were averaged for all mowers.

R. Brock:

Dave, do you have any idea of the variations you would expect in your results if you switched components but kept the same mower, for example, if you changed transmissions keeping the same type would you expect to get similar results or would you expect a large variation?

D.R. Tree:

No, I don't think you would get much variation. For example, if you switched to a different hydrostatic unit you would get shifts in peaks, but I think the engine would still be the dominant noise source.

D. Lyon:

First of all I didn't understand the reference on the power levels.

D.R. Tree:

That's because they would not let me give them to you.

D. Lyon:

Well, the other question I have is of more general interest: what is it about the University that makes this work pertinent to it compared to doing similar work in industry itself? Did you do this work here because the sponsor did not have the facilities or is there something unique about this kind of work that makes it fit the University?

D.R. Tree:

Well, the uniqueness of this work as far as we are concerned is the uniqueness of the Ray W. Herrick Laboratories in the fact that we do a lot of work for industrial people. One of these two companies had the facilities but not the manpower and they thought the work could be done better at the Laboratories rather than having to hire the manpower to do it. So it wasn't the uniqueness of the work, many industries could do it, it was the uniqueness of the Laboratories that we do this type of work for industry.

C. Hurst:

Could you give us a quick rundown on the type of mufflers on these engines? Were they more sophisticated?

D.R. Tree:

Yes, these are very simple expansion-type mufflers.

PUNCH PRESS NOISE RESEARCH, DEVELOPMENT AND DEMONSTRATION

J. R. Bailey
J. A. Daggerhart
Center for Acoustical Studies
North Carolina State University
Raleigh, North Carolina

INTRODUCTION

The Center for Acoustical Studies (CAS) became involved in punch press noise control following inquiries by a number of companies from the metal-forming industry. After initial studies and review of the literature, it was apparent that noise problems in metalforming are formidable and ready solutions are not available. Only modest progress has been made in quieting metalforming machinery despite readily acknowledged high noise levels. This may be attributed in part to the difficulties encountered in measuring and analyzing impact noise and vibration. The extensive fragmentation of the industry also has contributed to the dearth of information on noise control.

Research on metalforming machinery noise analysis began in the summer of 1973 with a pilot study supported by a major manufacturing company. Noise characteristics of several selected punch presses were defined, and a number of noise sources were identified. Techniques were developed for impact noise analysis, and a systems approach was developed for use in evaluation of noise control techniques. The result was a flow diagram which can be used for evaluating the feasibility of noise control through engineering.

Following the pilot study, the National Institute for Occupational Safety and Health (NIOSH) provided support for a major punch press noise reduction demonstration. This research was conducted over a sixteen month period in the plant of the aforementioned company. Usual procedure was to record sound and vibration data from plant tests for subsequent analysis in the CAS laboratories. Machine access, materials for punching, tool and die work, machine operators, and technical assistance from four members of the engineering staff were provided by the cooperating company.

At present the CAS is conducting a follow-up project aimed at optimization of tooling for minimum fracture noise in punching. Tool and die design as well as metallurgical aspects of punching are being considered in this effort.

Results of punch press noise control research are summarized in the following sections. In addition,

levels of effort are described in terms of manpower and monetary support.

RESULTS

In the majority of punch press operations, the primary noise sources may generally be grouped as 1) air exhausts, 2) mechanical impacts during the punching cycle, and 3) handling of finished parts. A typical noise signature of a single-cycle operation of an open-back, inclinable press is shown in Figure 1. The various events that occur during a cycle may be identified as 1) clutch actuation, 2) impact of the ram on the stripper, 3) impact of the stripper on the workpiece, 4) punching of the workpiece, and 5) actuation of the brake.

For the purposes of this paper, only the air exhausts and mechanical impacts will be treated. The control of the noise due to the handling of finished parts can, in most cases, be handled by relatively simple techniques.

AIR EXHAUSTS

Depending on the particular operation, a dominant source of noise during single-cycle operation of a punch press may be the air exhaust associated with pneumatic operation of the clutch and brake.

To provide a data base for the evaluation of several commercially available exhaust mufflers, tests were conducted in the North Carolina State University anechoic facility. During these tests, the impulsive sound level was measured as a function of the initial exhaust pressure. In addition, the time required for each muffler to exhaust an 0.001 cubic meter volume was monitored. The exhaust time data were necessary to insure that a particular muffler did not unduly restrict the exhaust and thus pose a safety hazard by altering the cycle time of the braking system.

The types of mufflers tested were 1) porous bronze, 2) radial flow, and 3) porous plastic.

Typical test results indicate that, at pressures near 90 psig, a noise reduction of approximately 20 to 35 dB may be realized depending on the particular type of muffler selected. It should be noted that

the reduction in a plant environment may not be as dramatic as that attained in a laboratory situation due to other sources that may be present. In fact, an evaluation of several mufflers in a plant operation yielded a maximum noise reduction of only 10 dB.

For air ejector systems used in parts handling, an unmuffled air jet can generate impulsive sound levels of 110 to 125 dB. Commercial silencers for air ejector systems can reduce this impulsive sound level by more than 20 dB at a typical system operating pressure of 90 psig. When the silencer is properly positioned, the force at the workpiece is at least equivalent to that produced by an open jet.

MECHANICAL IMPACTS

The analysis of typical punch press signatures indicates that there are many cases in which the dominant source is the actual punching operation.

During the punching operation, both the machine components and the workpiece will be subjected to simultaneous impulsive forces which will result in sound radiation from the ensuing vibrations. Test results have shown that direct sound radiation from the workpiece can be dominant when the dimensions of the workpiece are equal to or greater than the wavelength of radiated sound. From a large number of frequency analyses it has been observed that the sound spectrum tends to peak near 500 Hz. Thus, when characteristic workpiece dimensions exceed about four inches (10.2 cm), significant sound radiation can be expected from the workpiece.

An experiment was conceived to utilize the foregoing information to investigate the relative contributions of vibrations of machine components and the workpiece in sound radiation during punching. This experiment consisted of a series of tests in which only the area of the workpiece was varied. Theoretically, this variation should have had no effect on the reaction forces on the machine and thus little effect on the noise radiated by machine components. On the other hand, the noise radiated by the workpiece should increase with an increase in area. Results are shown in Figure 2, which is a plot of stripper impact and punching noise as a function of workpiece area.

It can be seen that for workpiece areas less than 25 square inches (161.3 square centimeters), the peak sound level is essentially constant. This is as expected from consideration of the radiation efficiency of the workpiece since the characteristic length for these sizes is less than 5 inches (12.7 cm). On the other hand, as the workpiece area is increased from 25 to 100 square inches (161.3 to 645.2 square centimeters), the peak sound level increases by an average of 4 dB. This result is as would be expected if vibrations of the workpiece and machine components contributed equally to the sound radiation.

From these results, it must be concluded that when the workpiece dimensions exceed the wavelength of radiated sound, both workpiece and machine component

vibrations contribute significantly to the radiated sound pressure level during the punching operation.

Some insight into understanding of the above observations can be gained by considering the response of simple systems to transient excitation. For a single-degree-of-freedom system, it can be shown that when the ratio of the pulse duration to the system natural period is much less than one, the maximum response of the system occurs after the force has dropped to zero. In this case, the responses can be reduced by increasing the mass of the structure. On the other hand, when the ratio of the pulse duration to the system natural period is much greater than one (i.e., the force is applied slowly), the maximum response occurs while the force is acting. In this latter case, the response is inversely proportional to stiffness, i.e., increasing the stiffness should reduce response and hence reduce noise. When the duration of the force is equal to one-half the natural period of the system, a pseudo-resonance exists. Control of resonant response can be achieved by detuning the system or adding damping. However a tenfold increase in the fraction of critical damping results in a decrease in maximum response of only about nine per cent. Thus, damping has little potential for reducing punch press noise.

The importance of these considerations of punch press noise cannot be overemphasized as they bear directly on the possible methods of controlling the vibratory motions during punch press operations.

For example, when workpiece areas are small, as in piece-parts operations, the vibrations of punch press components can be expected to be the dominant sources of noise. Due to the characteristic duration of the punching part of the machine cycle, the response of the structure will be controlled primarily by inertia effects. A substantial increase in structural mass is required to achieve a moderate reduction in system response. Therefore, in operations with small workpieces, it appears necessary to utilize a press of substantially higher tonnage capacity than would be called for on the basis of the forming operation. The use of a large capacity press is required to provide the massive structure necessary for control of structural vibrations.

When the workpiece size increases so that both workpiece and machine component vibrations are significant contributors to the radiated sound, significant noise reduction can be achieved only by controlling both sources. In most production situations it will not be economically feasible to constrain the workpiece or to substantially increase the mass of the press itself. In this case, the only viable alternative for noise reduction appears to be in the area of reducing the force levels involved during the punching operation.

For a given operation, force reduction can be accomplished by 1) reducing the punch-die clearance or 2) using shear ground punches. In Figure 3, peak noise levels are plotted for a flat punch operated over a wide range of clearances. It can be seen that above eight per cent clearance, the peak sound pressure level increases dramatically with

increasing values of the clearance. Thus, a reduction in the punch-die clearance is a relatively simple method of reducing the peak noise level associated with a given punching operation. It should be noted, however, that reductions in clearance can lead to degradation of the hole quality. The maximum clearance reduction for any given operation thus becomes a trade-off between the expected noise reduction and the possible decrease in hole quality.

Force reduction may also be obtained by the use of shear ground punches. In this case, the shear punches typically can be operated at a higher clearance value than a flat punch, giving a higher quality hole, while still operating with a lower value of the peak pressure level.

A comparison between the peak sound pressure levels associated with a flat punch, a female shear punch, and a male shear punch is shown in Figure 4. The geometries of the male and female shear punches are shown in Figure 5.

It may be seen that the female shear punch provides a reduction in the peak level for all clearance values tested as compared to the flat punch. The male shear punch is essentially the same as the female punch over the range of clearance values tested.

Further, the depth of the shear, D , had little influence on noise, i.e., an increase in D did not provide a corresponding decrease in the peak sound pressure level.

SUMMARY OF PROJECT ACTIVITY

Table I gives a summary of projects, support periods, personnel, support sources, and an estimate of total resources committed to punch press noise control from 1973 to the present. An estimate of manpower involved in punch press noise control activities is shown in Figure 6. It is estimated that approximately 5.5 man years have been committed to punch press noise. Monetary support has totaled \$72,672 since 1973.



Fig. 1 Typical punch press noise signature. Events are: (1) clutch engagement, (2) ram impact with punch, (3) punch impact with material, (4) material fracture, (5) brake activation

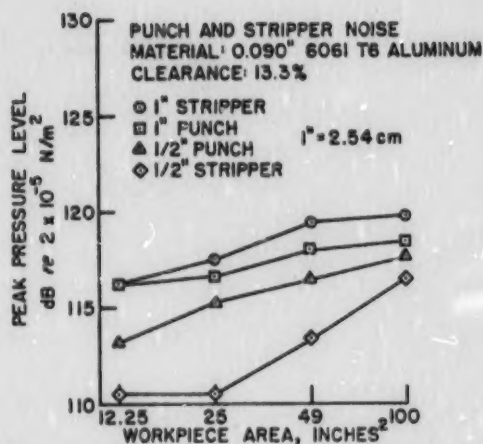


Fig. 2 Peak sound pressure level versus workpiece area

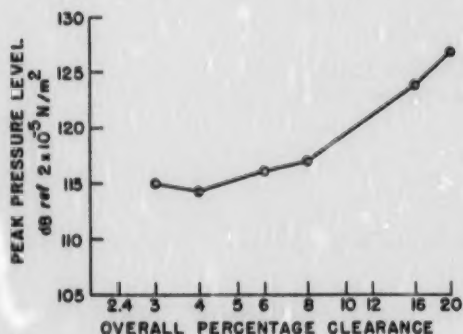


Fig. 3 Effect of tool clearance on punching noise

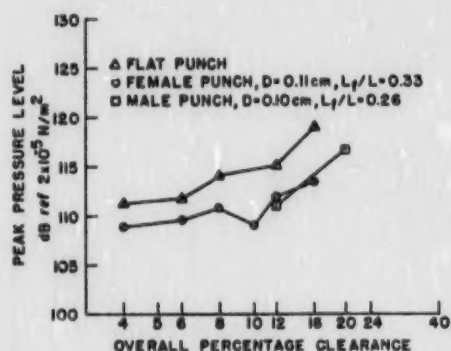
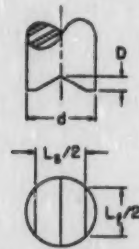
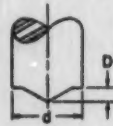


Fig. 4 Noise reduction due to use of shear punches

FEMALE SHEAR PUNCH

MALE SHEAR PUNCH



NOTE: $L_s/2$ AND $L_t/2$ ARE CIRCUMFERENTIAL MEASUREMENTS

Fig. 5 Diagrams of male and female shear puncher

Table 1 Summary of Punch Press Noise Control Activities

Project Title	Support Period	Personnel	Support Sources	Amount
Metalforming Machinery Noise Analysis	8/73 - 4/74	J. A. Daggerhart	Industry	\$ 8,857
		J. R. Bailey		
		D. Ferioli		
		T. Little		
		R. Magers		
		J. Jarrett		
Punch Press Noise Reduction Demonstration	4/74 - 9/75	J. A. Daggerhart	NIOSH	\$ 53,815
		J. R. Bailey		
		A. Akay		
		R. Carden		
		N. Stewart		
		R. Magers		
Punch Press Noise Control	4/75 - Present	J. A. Daggerhart	DuPont	\$ 10,000
		R. Cotter		

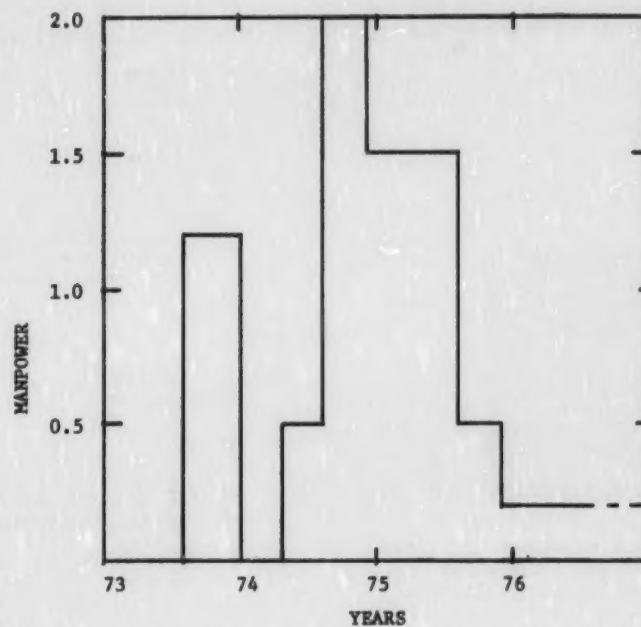


Fig. 6 Estimate of manpower committed to punch press noise control

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"Punch Press Noise Research, Development and Demonstration"
presented by
J.R. Bailey

DISCUSSION:

S. Waggoner:

What kinds of noise reductions did you get in industry? Can the operator not wear earmuffs yet?

J.R. Bailey:

Well, the regulation we were seeking to meet was the 120 dB peak level, and we were able to achieve that for the operations we considered. There is something like 480 thousand shears and punch presses each capable of doing dozens of different jobs so it becomes almost an intractable problem unless you set yourself some bounds. So, we chose a systematic approach looking at a large number of machines, narrowing it down to one and then looked at a number of operations. I certainly cannot make a blanket statement that all punch presses can be controlled. There is so much work to be done!

P. Mansbach:

Where were these sound pressure levels measured?

J.R. Bailey:

These were measured at approximately ear level just to one side of the operator's normal position. We did take some measurements around the machine and found very little variation. We were in the near field in all cases.

P. Baader:

We did some such studies at Carrier years ago and there we had a very peculiar punch press to work with. The material was very soft, so there was no impact of the work piece on the material; no fracture noise at all; and the clutch was well designed so it was quiet. But what happened was that this thing had a traveling pad which came down and in coming up and getting into position for the next stroke, after all the work was done, it hit a keeper (clap!) that is the kind of noise which is totally unnecessary. If you're lucky, you'll find one of those and you can be a hero!

D. Muster:

How do you define percentage clearance? Is the thickness involved?

J.R. Bailey:

Yes, percentage clearance is defined as the diameter of the die minus the diameter of the punch divided by the thickness of the material times a hundred.

NOISE REDUCTION OF A COLD-HEADING MACHINE

Lynn L. Faulkner, Associate Professor
Glen C. Steyer, Graduate Research Assistant

Department of Mechanical Engineering
The Ohio State University
206 W. 18th Avenue
Columbus, Ohio

INTRODUCTION

Many metal forming and shaping operations such as forging, cold-heading and stamping utilize mechanical impact tools on the work piece. While considerable progress has been made in reduction of certain types of machinery generated noise in the past few years, comparatively little has been reported on actual noise reduction of the sound generated by impacting type machinery. The usual initial assumptions are that the impact process of forming the part is the mechanism responsible for the production of sound energy for this type of machinery. Antochow (2) utilized a simple impact machine for study of the sound generated by the acceleration of the air between the impacting parts termed "slap" noise. The results of his analysis showed that sound can be produced before actual metal-to-metal contact of tool and part depending upon the geometries involved. Nishimura and Takahashi (7) investigated the acceleration (or deceleration) of air in the direct impact of steel spheres as an impact noise generation phenomenon. Abrishaman (1) studied the parameters related to the sound radiated from stress waves in rods struck by steel balls or rods. The effect of impact velocity and workpiece material has been reported by Bruce (3). Multiple impacts on work-hardening material can result in a change in sound level with successive blows of the forming process. Even for materials which are not work-hardening there is evidence that successive forming blows produce greater sound pressure levels. One explanation is that the first impact in part forming usually deforms a greater volume of material than successive impacts and hence more energy is absorbed in the part for the initial impact. Final impacts are typically utilized to achieve surface finish and dimensional requirements which deform small volumes of part material and absorb little energy.

An occurring result of many impact machine noise studies is the sound generation of

the so-called "conventional" machine elements; sometimes to the extent of masking the direct impact noise. Patterson (8) reported the presence of all conventional machine element noise sources such as gears, bearings, slides, cams, etc., on a cold-heading machine. The excitation of panels by impact was noted. Measurements were made of the sound levels with diagnostic studies to establish contribution of direct impact, air jets, and cover vibration of sound level.

MACHINE DESCRIPTION

This paper reports an analysis of a high speed, double blow, cold header machine as shown in Figure 1. This machine produces 1/8 inch button head rivets at a rate of 467 per minute. The stock material is 1/8 inch SAE 1016 recessed quality header wire. There are several stages to the rivet forming operation. First, an appropriate length of wire is cut. A set of transfer fingers position the blank in front of the die station. A cone punch then pushes the blank into the die and performs the initial forming operation. The second impact is by a finish punch which forms the finished rivet head, after which, the formed rivet is ejected from the die by means of a push rod and an air jet ejector. The cut-off lever, transfer fingers, and punch assembly are all driven from a single camshaft. The kickout utilizes a rod which is driven forward by a cam to push the finished rivet from the die. Once the finished rivet is pushed out of the die, it is blown into a collecting chute by means of an air jet. The air jet also simultaneously lubricates the punch area with a mist oiler system attached to the air supply system. As can be seen, this machine utilizes many potential noise producing elements such as cams, gears, bearings, and air jets in addition to the impact mechanisms in the rivet head formation.

MEASUREMENTS

Sound pressure levels were measured with a 1/2 inch Bruel and Kjaer condensor microphone. Frequency analysis was performed with a 2107 Bruel and Kjaer constant percentage narrow band analyzer and with a UA-500 Federal Scientific Ubiquitous Real Time Analyzer. All sound measurements were made in an anechoic chamber with an 80-hertz lower cutoff frequency. This chamber is part of the acoustic facilities at The Ohio State University and was constructed with four mounting posts in the center of the floor area to support heavy machinery such as the cold-header of this study.

Acceleration measurements were made on machine panels to determine both frequency content of the panel motion and the amplitudes of the cover panel motions. Acceleration signals were analyzed in a similar manner as the sound pressure signals described above. Cam displacement profile and transfer mechanism displacement motions were directly measured by utilizing a rotational potentiometer attached to the flywheel and linear potentiometers attached to the translating components. The rotational potentiometer was connected to the X-axis and the linear potentiometer to the Y-axis of an X-Y plotter. As the machine was rotated slowly through a cycle a plot of component displacement could be obtained. Since the flywheel rotates at a constant velocity for this slowly rotating case, the result is an X-axis displacement which can be related to time. The result is a displacement versus time signal for analysis of frequency content.

NOISE SOURCE IDENTIFICATION

Three procedures were utilized in this study to identify and characterize the noise sources and rank them in order of importance. The first method was to disconnect various components in the machine one at a time and measure the resulting sound levels. This by necessity dictated that levels with and without stock in the machine had to be established since removal of components prevented actual forming of rivets. This procedure was useful in identifying sources and ranking the sources in order of decreasing sound level production.

A second method of source analysis was to compare the frequency content of the sound pressure with the frequency content of sources. This was done by frequency analysis of microphone measurements in the near acoustic field of various sources, acceleration measurements on various components such as cover panels, frequency analysis of various cam displacements and frequency analysis of transfer motions of various mechanical components from rotational and linear potentiometer signals. Measurements

made at various positions around the machine for components not operating were used to classify and rank the machine components and operations in regard to noise generation potential. A typical frequency spectrum is shown in Figure 2.

A third method of source identification was to display the sound pressure level versus time trace on a dual beam oscilloscope with a timing trace on the other beam. This timing trace was obtained from the flywheel or other known reference signal from the machine.

Some of the noise source contributions were easily identified by the linear or A-weighted measured sound pressure levels. Other sources could only be identified by comparison of frequency spectrums and a visual correlation of sound pressure level frequency response compared to acceleration or displacement frequency response data.

RESULTS OF MEASUREMENTS

The first measurements were taken for the machine in normal operation making rivets and for the same conditions not making rivets. The result was that the A-weighted sound pressure level was unchanged when no stock was fed to the machine. This implied that the impact generated sound level was less important than the other sources.

The most easily identified source was the air ejection of finished parts from the machine. Elimination of the air jets reduced the sound level approximately 1 dBA.

A reduction of 4 dBA resulted when the transfer mechanism was disconnected. This mechanism transfers the blank cut from the stock feed wire to the forming die. Because of the rapid machine operation, the transfer mechanism is required to position the blank and rapidly retract so that the cone punch will be cleared. This results in rapid displacements of the mechanism and hence large accelerations. A plot of transfer mechanism motion versus cam angle is shown in Figure 3. A frequency analysis revealed many harmonic components of this motion. Sound is apparently generated due to reaction forces on the main frame of the machine as a result of the inertia forces of this transfer device. The device itself is quite rigid and small, therefore direct sound radiation is discounted.

The next most significant noise source was a punch rocker stop. This machine requires two punch operations for one rivet; a cone punch and a finish blow. This is achieved by one slider with a rocking head which contains two tools. To accurately position the rocking head an impact stop is employed as shown in Figure 4. This impacting of the head against the stop accounted for an

additional 2 dBA sound reduction after elimination of the transfer mechanism.

The fourth source identified after elimination of the above was the mechanical "kick out" which ejected the finished part from the die. This operation was found to result in an additional 1 dBA noise reduction.

Other sources identified were two panels over the cam shaft and over the top portion of the machine. For the complete machine making rivets, a reduction of slightly more than 1 dBA could be realized by adding damping material to these panels.

The result of elimination of all of these sources simultaneously resulted in a combined noise reduction of 7 dBA.

PROPOSED MACHINE MODIFICATIONS

After identifying the noise sources, the following modifications were suggested ranked in order in which they must be implemented.

1. Eliminate the air part ejection system and eject parts with a high velocity oil system.
2. Redesign the blank transfer mechanism so that the acceleration can be reduced.
3. Provide an energy absorbing, slowly decelerating stop for the punch rocker head assembly to eliminate impact.
4. Eliminate metal-to-metal impact between kick-out cam and kick-out rod.
5. Apply damping material to certain panels and covers.

Item 1 above was implemented on the test machine and was found satisfactory. The machine was equipped with an oil pump for lubrication therefore minor changes were necessary to utilize an oil ejection system to replace the air ejector. Item 2 requires a redesign of the parts and cam. This modification was not made on the test machine but is planned for a future prototype. This source must be reduced before the following items will be effective. Items 3, 4 and 5 are all possible with minor modification to the machine. Some development is required to assure satisfactory machine operation with these changes. The indication from this study is that the A-weighted sound pressure level can be reduced to well below 85 dBA, with a series of these machines side by side the resulted level is calculated to be below 85 dBA.

ACKNOWLEDGEMENT

This work was completed in nine months utilizing 8 man-months of effort and was supported by grants from the National Machinery Company, Tiffin, Ohio totaling \$8000.00.

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Figure 1. Photograph of the cold-heading machine studied.

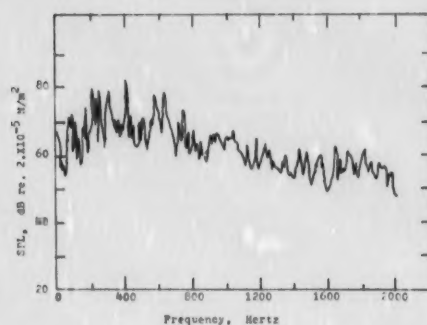


Figure 2. Typical frequency spectrum of the operator location sound pressure level.

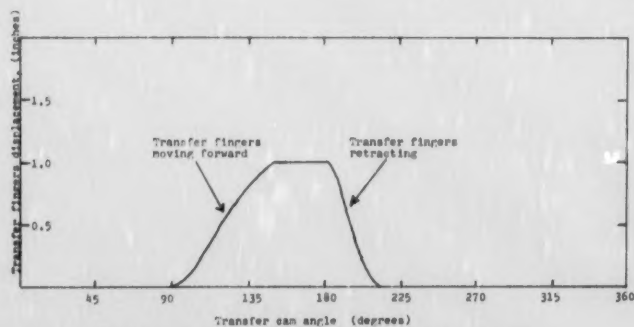


Figure 3. Profile of the blank transfer mechanism displacement.

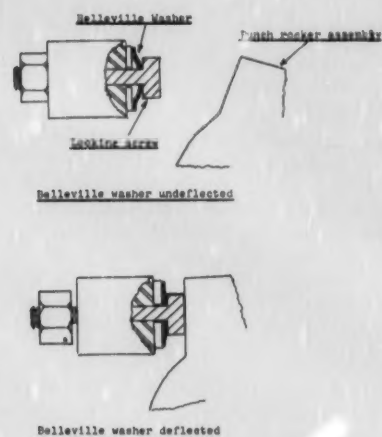


Figure 4. Diagram of the punch rocker head assembly impact stop for accurate positioning of the tool.

"Noise Reduction of A Cold-Heading Machine"
presented by
L.L. Faulkner

DISCUSSION:

D. Muster:

Your Table 3: I just don't believe it! There is just no way I can persuade myself that you can have a 40th harmonic and have it mean anything when your talking about acceleration. Admittedly these are computer generated, but how much did you actually use?

L.L. Faulkner:

We used only the lower harmonics.

D. Muster:

Well, you should not include the rest, it's garbage, and it tends to mislead people who read it.

W. Gatley:

In the blank transfer mechanism, did you find radiation from the mechanism itself or merely the fact that it excited the structure around it?

L.L. Faulkner:

In itself it did not have significant radiation, but the forces acting back thru the cam and bearings were exciting the frame and castings.

PREVENTING ABNORMAL COMBUSTION NOISE

Peter K. Baade
Visiting Professor

Ray W. Herrick Laboratories
School of Mechanical Engineering
Purdue University
West Lafayette, Indiana 47907

NEED FOR RESEARCH ON COMBUSTION OSCILLATION

Oscillatory combustion is a source of abnormal combustion noise which can occur in virtually any type of combustion systems ranging from furnaces for residential heating systems with an output as low as 60,000 Btu/hr. and combustion chambers measuring only 2 ft. to blast furnace stoves with combustion chambers standing 100 ft. tall. Typically, self-excited oscillations will only occur under some operating conditions. In small devices, such as furnaces, they may occur only briefly during start-up; but even this is totally unacceptable because of the annoyance of the radiated noise. In large systems, such as blast furnace stoves, there is not only annoyance, but sound pressure of sufficient magnitude to destroy the brickwork and causing physiological damage.

Combustion engineers typically deal with these problems in an empirical manner if and when they arise. At times when there was little pressure for technological innovation, such empirical approaches may have been adequate. During periods requiring rapid innovations, however, the empirical approaches are not satisfactory. Periods of rapid innovation are typically brought about by changes in the fuel supply situation. One such period occurred in the United States in the late 1940's and early 1950's when natural gas replaced manufactured gas for industrial and domestic use. During this period all existing installations had to be changed over and many new designs evolved to meet the requirements of the rapidly expanding utilization of gas. These radical changes in utilization and design led to many occurrences of self-excited oscillations which, in turn, provided the motivation for an increased research activity aimed at providing a better understanding and more rational approaches to preventing such oscillations. A similar situation occurred in Europe in the 1960's after natural gas became available in the areas surrounding the North Sea.

In the United States we have enjoyed a period of 20 years during which the occurrence of combustion oscillations has been relatively rare because the rate of change in design and utilization has been quite slow. This period is about over: Shortages in the supply of natural gas make it necessary to use

alternative gases for peak shaving and as a standby fuel. The introduction of peak shaving gas by utilities has caused combustion oscillations in a number of furnaces and boilers which have operated satisfactorily on natural gas for many years. New installations and designs must be able to operate satisfactorily on several different fuels.

Another emerging pressure for design innovation is the need to conserve gas by achieving higher efficiencies. This will require departures from well proven designs. In this situation there is a clear need to intensify research efforts to fill-in the gaps which still exist in our knowledge of combustion oscillations in order to develop rational design procedures for meeting the new requirements without risking noise pollution due to combustion oscillation. Compared to earlier research efforts, we are fortunate in having at our disposal vastly more powerful instrumentation for experimental analysis as well as more powerful tools for mathematical analysis and modeling.

Combustion oscillation is, of course, not the only type of combustion noise affecting the environment around a furnace, boiler plant, or other combustion device. The other types of noise, commonly referred to as combustion roar or hiss, are caused, however, by entirely different mechanisms and the methods of controlling them are, therefore, entirely different from the methods required for controlling combustion oscillation. The research project at Purdue University on which this paper is based dealt exclusively with the control of combustion oscillations.

CHARACTERISTICS OF COMBUSTION OSCILLATION

Inasmuch as the methods required for controlling combustion oscillations are entirely different from those required for combustion roar and combustion hiss, it behooves us to review the distinguishing characteristics. Combustion oscillations are self-excited. The spectrum of the radiated noise is, therefore, dominated by a single frequency often associated with side-bands and/or higher harmonics of that frequency. Combustion roar and combustion hiss, on the other hand, are random phenomena and, therefore, have a broadband spectrum. This broad-

band spectrum may have a concentration of energy in a certain frequency range, but never at a discrete frequency. In approaching an existing noise problem, it is important, therefore, to start with a spectrum analysis of the noise in order to select the most appropriate method of control.

Another distinguishing characteristic is that combustion oscillations in a given installation usually occur only at certain operating conditions (firing rate, fuel-to-air ratio, fuel composition, etc.) but are totally absent at other operating conditions. It is not uncommon for combustion oscillations to occur during a brief interval at start-up and then cease abruptly, but the opposite can also happen. In either case the difference between presence and absence of combustion oscillations typically amounts to well over 20 dB. Control of combustion oscillations, thus, means not merely reduction but rather total elimination of the oscillatory mode.

The objective of the research project at Purdue University has been to provide practical information which can be applied at the design stage to prevent the occurrence of combustion oscillation in a proposed device or installation. To this end, stability criteria were formulated for the feedback loop which is responsible for exciting oscillations. Application of these criteria requires a knowledge of the amplifying characteristics of the flame. A method was, therefore, developed for measuring the transfer function of aerated, fuel rich flames. Such flames are used in many types of practical combustion devices. The data obtained were the first for such flames to appear in the literature. It was also recognized that dynamic stability analysis of feedback loops and transfer functions are unfamiliar concepts for most people in the heating industry. A special effort was, therefore, made to generate information that would help communicate the results of the research effort to industry.

A typical example of this effort is the production of a short movie which shows in slow motion the self-excited oscillations of the flame on the multi-port burner used in this investigation in a simple, transparent combustion chamber. This movie demonstrates that there are two separate but inter-related phenomena that occur during combustion oscillation:

1. Oscillations of the burning rate of the flame. In this particular case (and in many others) these manifest themselves in visible oscillations of the size and position of the reaction zone.
2. Oscillations of the pressure in the combustion chamber.

These two oscillations occur at exactly the same frequency with a fixed phase relationship to each other. The pressure oscillations are the cause of the flame oscillations, but they are also caused by the flame oscillations. Thus, the cause-effect relationships between these two phenomena form a closed loop as shown in Fig. 1. Recognition of the existence of such a closed loop and an understanding of the manner in which this leads to self-excitation

are the key to any rational solution and ultimately prevention of combustion oscillation problems and the noise pollution associated with them.

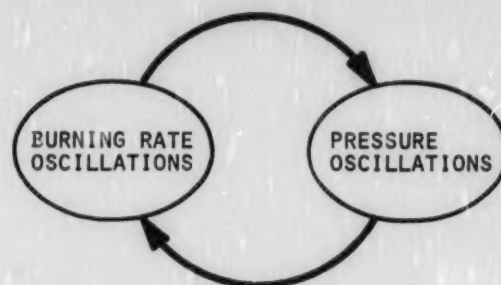


Fig. 1 Cause-effect Relationships in Oscillatory Combustion

CRITERIA FOR COMBUSTION OSCILLATIONS

The existence of this closed loop of causes and effects and its role in self-excited combustion oscillations was recognized and very lucidly described by Rayleigh⁽¹⁾ almost 100 years ago. Rayleigh pointed out that the occurrence of such oscillations is critically dependent on the magnitude and, particularly, the phase of the combustion rate oscillations relative to the pressure oscillations which cause them. He also points out that this relationship, which is represented by the arrow in the bottom part of Fig. 1, depends on the dynamic response lag of the flame itself and on the length of the fuel supply line relative to the wavelength at the frequency of oscillation. He summarizes the importance of the phase angle of this relationship by stating that oscillations are most encouraged when the resulting oscillations in the heat release rate of the flame are in-phase with the pressure oscillations which cause them, and that self-excited oscillations cannot occur when this phase relationship is near 180° instead of 0°.

These conditions have been used to good advantage by all of the early investigators of self-excited combustion oscillation problems and are commonly called the "Rayleigh Criterion".

The most extensive and fruitful use of the Rayleigh Criterion has been made by Abbott Putnam and his co-workers at Battelle. The result of their investigations are summarized in Putnam's book: "Combustion Driven Oscillations in Industry"⁽²⁾, which describes occurrences and solutions of combustion oscillation problems in a large number of different combustion systems.

Putnam has expanded the Rayleigh Criterion and put it on a quantitative basis by reasoning, from energy considerations, that the oscillations can build up and maintain themselves only if the following⁽³⁾

inequality condition is met:

$$\int_0^T h \cdot p \, dt > 0 \quad \text{Eq. (1)}$$

where T is the period of the oscillation,
 h is the time varying heat release of the flame, and
 p is the time varying pressure in the combustion chamber at the location of the flame.

The left-hand side of Eq. (1) is a quantity proportional to acoustic energy generation per cycle. Putnam pointed out repeatedly that this rate of energy generation does not only have to be positive rather than negative, but it has to be equal to or larger than the rate at which acoustic energy is dissipated from the system.

In the 20 years since most of Putnam's work was done, many investigators of combustion oscillations have formulated different criteria based on transfer function considerations. So far, the transfer function approach has found little application in industry; and the Rayleigh Criterion is still extensively used. Since the transfer function approach is much more powerful, this approach has been used in the work at Purdue University; and it has, indeed, been found to be very fruitful. To help its acceptance in industry I would like to show that Eq. (1) can be recast into a form which will lead to the same criterion as the transfer function approach. For this purpose, the left-hand side of Eq. (1) is reformulated so that it represents acoustic energy generation per unit of time rather than being merely proportional to that energy generation. The right-hand side is reformulated to represent acoustic energy dissipation as follows:

$$\frac{1}{T} \int_0^T q \cdot p \, dt \geq \langle p^2 \rangle \operatorname{Re} \left(\frac{1}{Z} \right) \quad \text{Eq. (2)}$$

where q is the time varying volume expansion rate of the flame,
 $\langle p^2 \rangle$ is the mean square value of the pressure at the flame, and

$\operatorname{Re} \left(\frac{1}{Z} \right)$ is the real part of the acoustic admittance of the combustion chamber, at the flame location, at the frequency of oscillation.

If the left-hand part of Eq. (2) is equal to the right-hand part, oscillations will maintain themselves but neither grow nor decay. If the left-hand side is larger than the right-hand side, the oscillations will grow.

Since Eq. (2) needs to be evaluated at a specific frequency (the frequency of oscillation), the following substitutions are in order:

$$p = |p| \cos (2 \pi f t) \quad \text{Eq. (3a)}$$

$$q = |q| \cos (2 \pi f t + \theta) \quad \text{Eq. (3b)}$$

$$\langle p^2 \rangle = \frac{1}{2} |p|^2 \quad \text{Eq. (3c)}$$

where $|p|$ is the amplitude of the pressure oscillation at the flame location,

$|q|$ is the amplitude of the volume expansion rate of the flame,
 θ is the phase angle of q relative to p , and
 f is the frequency of the oscillation
 $(f = \frac{1}{T})$.

With these substitutions Eq. (2) can be solved and rearranged as follows:

$$\frac{1}{2} |q| |p| \cos \theta \geq \frac{1}{2} |p|^2 \operatorname{Re} \left(\frac{1}{Z} \right) \quad \text{Eq. (4a)}$$

$$\frac{|q|}{|p|} \cos \theta \geq \operatorname{Re} \left(\frac{1}{Z} \right) \quad \text{Eq. (4b)}$$

$$\geq \frac{1}{|Z|} \cos \phi \quad \text{Eq. (4c)}$$

where θ is the phase angle of the oscillations of the volume expansion rate ("volume velocity") of the flame relative to the oscillations of the pressure, and
 ϕ is the phase angle of the pressure relative to the volume velocity.

Obviously, θ must be $-\phi$ so that $\cos \theta = \cos \phi$. The ratio of $|q|$ to $|p|$ in the left-hand side of Eq. (4b) is the magnitude of the relationship; (i.e., transfer function) represented by the bottom arrow in Fig. 1. In Reference 4 this relationship has been designated by the symbol $(H \cdot G)$. Thus, Eq. (4) can be rewritten as:

$$|H \cdot G \cdot Z| \geq 1 \quad \text{Eq. (5a)}$$

$$|Z| \geq \frac{1}{|H \cdot G|} \quad \text{Eq. (5b)}$$

Eq. (5a) and (5b) represent the "magnitude criterion" for the occurrence of self-excited combustion oscillations derived by transfer function analysis in Reference 4. It should be noticed that this criterion applies at any frequency at which the sum of the phase angles of Z and of $H \cdot G$ is zero. This condition was implied above by stating that $\theta = -\phi$ because θ is the phase angle of $H \cdot G$ and ϕ is the phase angle of Z .

It should be recognized that the phase criterion given above is that of a positive feedback loop. Several investigators (5, 6) have derived stability criteria from the properties of a negative feedback loop. This is entirely legitimate and merely a question of the sign conventions used in defining the symbols. The symbol Y used in Reference 5 is, for instance, related to the symbol H used in Reference 4 by: $H = -Y$.

APPLICATION OF STABILITY CRITERIA TO THE DESIGN OF COMBUSTION CHAMBERS AND SUPPLY LINES

Ideally, the designer of a combustion system should have quantitative information on the magnitude and phase of the three transfer functions Z , H , and G as functions of frequency. While some data can be found in References 5 - 13, complete information is not, at the present state of the art, generally

available. Certain qualitative conclusions can, however, be drawn from Eq. (5) and applied at the design stage to minimize the likelihood of combustion oscillations.

Eq. (5b) shows that oscillations cannot occur unless the magnitude of Z exceeds that of $\frac{1}{H \cdot G}$. Calculating Z for the very simple combustion chamber shown in Fig. 2 one gets the curves of the magnitude and phase angle shown in Fig. 3. The solid line for

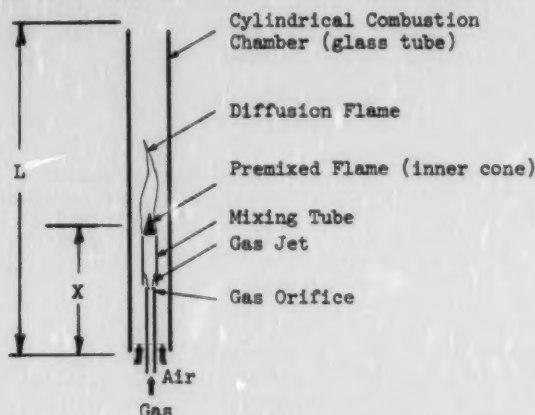


Fig. 2 Burner and Transparent Combustion Chamber for Demonstrating Effects of Combustion Chamber Design on Stability

small damping shows a sharp peak at $L/\lambda = 0.5$; that is at the lowest natural frequency of the cylindrical combustion chamber. Also shown in Fig. 3 is an estimate (dotted curve) of $\frac{1}{H \cdot G}$ consistent with the measurements made in our project at Purdue University. This curve is exceeded by the peak in the $|Z|$ curve for small damping but not by the lower peak in the broken curve for $|Z|$ pertaining to increased damping. In the design of combustion chambers it is, therefore, desirable to provide as much damping as practical.

It should be pointed out that the word "damping" is used here in its broadest sense. It is not restricted to devices which absorb acoustic energy, but also includes any design features which enhance the radiation of acoustic energy from the combustion chamber to the environment. At first glance, this may appear to be counter-productive to the goal of reducing noise radiated to the environment. It can be demonstrated, however, that this form of dissipation actually results in less noise radiation provided that there is enough of this dissipation to prevent build up of self-excited oscillations within the combustion chamber.

As far as the design of the supply lines is concerned, it should be pointed out that for many arrangements (not the one shown in Fig. 21)

$\frac{1}{H}$ is the negative impedance of the gas or mixture supply tube at the burner head. Thus, any design features which increase this impedance will reduce

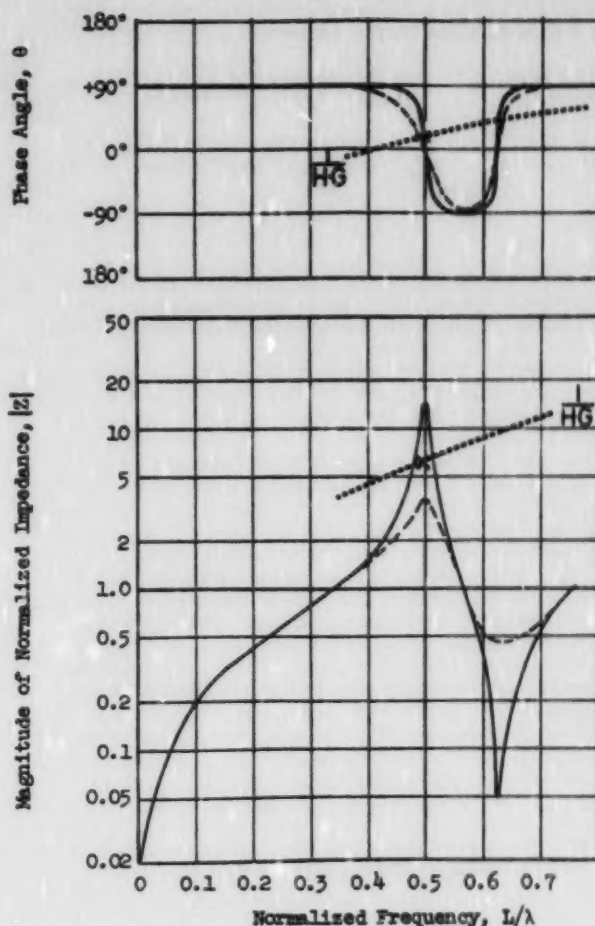


Fig. 3 Effect of Damping on Impedance of Combustion Chamber of Fig. 2 for $X/L = 0.2$ ($\lambda = \text{wavelength}$)

— Bare Combustion Chamber } Z
 --- Damping Added at Top

the likelihood of self-excited combustion oscillations. If this is done by optimizing the length of the supply line, as is common practice in the design of blast furnace stoves, it must be borne in mind that the effectiveness of this tuning depends on avoiding damping in the supply line (see Reference 9).

INTERPRETATION AND MEASUREMENT OF FLAME TRANSFER FUNCTIONS

In Eq. (5a) and (5b) G stands for the transfer function of the flame defined in such a manner that G is zero when the flame is not burning. Fig. 3 shows that this transfer function is important in two ways:

1. If G is small enough at the natural frequency of the combustion chamber where the curve for Z has its peak, then combustion oscillations cannot be self-excited regardless of the phase angle of G .
2. Even if the magnitude of the flame transfer function is such that the curve for $\frac{1}{H \cdot G}$ is exceeded by the curve for $|Z|$ at some frequencies, self-excited oscillations will not occur provided that the phase angle of $\frac{1}{H \cdot G}$ is not too close to 0° at such frequencies. For many burner configurations, the phase angle of H will be around $+90^\circ$ over a large frequency range. For such burners it is, therefore, desirable to avoid phase angles of G between -60° and -120° at the natural frequency of the combustion chamber.

A typical plot of the magnitude and phase angle of G for the multi-port burner used in the Purdue project is shown in Fig. 4. This set of data was taken with an air-to-fuel ratio of 19 which results in a very "hard" and compact flame (the primary air supply was 82% of the stoichiometric amount). This flame with its associated burner has the greatest tendency to go into self-excited oscillations if the natural frequency of the combustion chamber is between 80 Hz and 180 Hz because the phase angle of

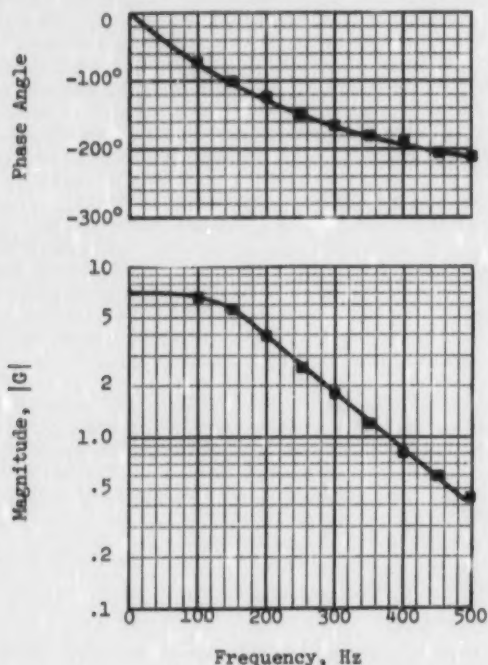


Fig. 4 Transfer Function of an Aerated Propane Flame on Multiport Meker Burner

the flame transfer function in this frequency range is between -60° and -120° and the magnitude is above 5. If the natural frequency of the combustion chamber is raised to 350 Hz or above, however, the phase relationship of the transfer function G would make it quite impossible to have self-excited oscillations. In actual practice it would not be necessary to raise the natural frequency quite that high because the combination of the increase in phase angle and the fall-off of the magnitude of G result in a rapid decrease in the likelihood of oscillation if the natural frequency of the combustion chamber exceeds 250 Hz. This can be demonstrated quite readily by using a combustion chamber similar to that shown in Fig. 2 but with a telescoping extension which permits changing the natural frequency of the combustion chamber.

The data shown in Fig. 4 were obtained with a novel measurement technique developed under the Purdue project and described in Reference 10. It consists of modulating the flow of the gas/air mixture to the burner head by means of a loudspeaker attached to the mixing tube and measuring the radiated sound pressure magnitude and phase angle at various frequencies, both with the flame "on" and with the flame "off". From these measurements the transfer function G is calculated as follows:

$$G = \frac{P_{on}}{P_{off}} - 1 \quad \text{Eq. (6)}$$

This measurement method is quite simple, but it is rather tedious because it involves measurements at many discrete frequencies. Particularly at high frequencies this method is also very sensitive to errors in phase measurement and to changes in the transmission of sound from the flame to the microphone brought about either by reflections in the measurement environment or by temperature effects when the flame is on. Errors in the measurement can be spotted by plotting the complex ratio of P_{on}/P_{off} on polar coordinate paper and repeating any measurements which do not agree with a smooth curve through the majority of the data points. From the resulting smooth curve one can obtain the transfer function by numerical means according to Eq. (6) or by graphical means by shifting the origin on the real axis by the amount of 1.0. The graphical procedure results in a Nyquist plot of the transfer function as shown in Fig. 5 for the same data points which were shown in Fig. 4.

RECOMMENDATION FOR THE APPLICATION OF FLAME TRANSFER FUNCTIONS TO BURNER DESIGN

In the preceding section, the significance of the magnitude and phase of the flame transfer function at the natural frequency of the combustion chamber was discussed. For a given burner and flame, self-excited combustion oscillations can, in principle, be prevented by designing the combustion chamber such that its natural frequency is above a value determined from the transfer functions of that burner and flame. There are, however,

practical limits to this. A better approach generally lies in designing the burner so as to obtain a flame which assures stability without undue design constraints for the combustion chamber. This requires information on the effect of the burner parameters on the transfer function of the flame.

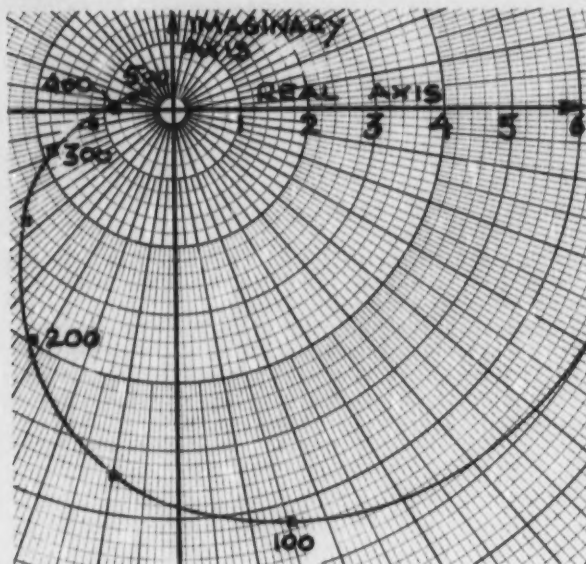


Fig. 5 Transfer Function of Fig. 4 in Nyquist Representation

In the project at Purdue University, flame transfer functions have been measured for a few burner port geometries at a number of different air-to-fuel ratios and port loadings. These measurements show that both the phase curve and the magnitude curve of Fig. 4 can be shifted to substantially lower frequencies by decreasing the air-to-fuel ratio (15, 16). This results in a "softer" flame having a larger inner cone and a marked decrease in the tendency to excite oscillations as discussed in some detail in References 5 and 10. Changes in the air-to-fuel ratio have long been recognized as a primary means for eliminating combustion oscillations in many existing systems. The measurements obtained provide at least a qualitative guidance for such efforts.

These measurements are consistent with the theoretical prediction (Reference 14) that the phase magnitude curves of G should shift to lower frequencies if the size of the pre-mixed portion of the flame is increased without increasing the mixture velocity; i.e., by increasing port size. The measurements do not confirm, however, the theoretical prediction of Reference 14 that the high frequency part of the magnitude curve should be inversely proportional to frequency. Instead, the measurements show a relationship inversely proportional to the square of the frequency. Also, the measured phase lags are much greater than those theoretically predicted in Reference 14.

These discrepancies are not surprising as the theoretical derivation of Reference 14 assumes that the shape of the oscillating flame remains conical (for a circular burner port), changing merely in height but remaining fixed at the base. This is not consistent with experimental observations (see Fig. 6). In the long term, it is certainly within the scope of the project started at Purdue to derive a more adequate theory for predicting flame transfer functions.

For the shorter term, it is proposed to obtain enough experimental data to establish empirically the influence of the common design variables in burner geometry and operation on flame transfer functions. This should be done for the various fuels which modern combustion systems have to be able to handle without abnormal noise emission. A generalized relationship to be derived from this data can be applied to optimize the design of burners to operate, as usually required, in a variety of combustion chambers. Specific data for a given burner design will be useful to check proposed applications of that burner in conjunction with experimental or general data on the combustion chamber.

During the three years during which the project at Purdue University was funded by the American Gas Association (AGA) with additional grants-in-aid from the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), a basis for the measurement and application of flame transfer functions was laid. During the next, as yet unfunded, phase of this work, some already identified refinements in the measurement techniques should be implemented and the appropriate data obtained for a representative set of burners and operating conditions. Liaison with industry is essential both for picking suitable burners to be tested and for translating this technology from the university to industry. ASHRAE has already appointed a liaison committee for this purpose. A demonstration of the application of flame transfer functions to a full scale combustion system is proposed. A furnace which is stable with natural gas but oscillates when operated on other fuels is already on hand for this demonstration.

As part of the next phase of this work, it is also proposed that the result of similar work going on in Europe be evaluated and utilized to reach the objectives of the project in a cost-effective manner. The necessary contacts have already been established.

ACKNOWLEDGEMENTS

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American Gas Association:
(3) Grants totaling \$50,000

American Society of Heating, Refrigerating and Air-Conditioning Engineers:

(2) Grants-in-aid totaling \$10,000

Plus services of the industry Liaison Committee consisting of representatives of various ASHRAE Technical Committees for reviewing and guiding the project.

Carrier Corporation, Syracuse, New York:

Furnace and burner hardware, test equipment, and data.

Co-principal investigators for this investigation are Professors V. Goldschmidt and R. Leonard of the Ray W. Herrick Laboratories, Purdue University.

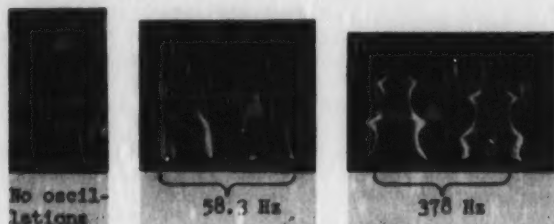


Fig. 8 Instantaneous Flame Shapes of Premixed Portions of Propane-Air Flame on 0.5 in. Dia. Port at an Air-to-Fuel Ratio of 12.5 (from Reference 17).

Note: Stroboscopic pictures at oscillating frequencies of 58.3 and 378 Hz are taken half a period apart.

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"Preventing Abnormal Combustion Noise"
presented by
P. K. Baade

DISCUSSION:

R. Cohen:

I thought I should just point out for the record that this was not structural damping, but damping in the acoustical mode.

P. Baade:

That's right.

R. Lambert:

Can you calculate the critical damping in a flue for a typical installation in advance?

P. Baade:

No, you can't because you do not have quantitative information on the transfer functions for anyone of the three elements at this point. If you had a transfer function of the flame; and transfer function for the fuel supply, H; and if you could model the transfer function, Z; which does contain damping as one of the parameters; then, yes, you could calculate the whole shebang.

D. Muster:

In going from equation 4 to 4b, how did you do it?

P. Baade:

I divided the right hand and the left hand by $|p|^{2/2}$.

K. McConnell:

You have a simple model here and you showed us a stack, but what happens when you have a big burner in there and there's a lot of piping down the tube and you have cyclone separators and all sorts of magnificent impedances? How do you control one of those? I'm aware of one that's marginally stable and every once in a while it gets out of hand.

P. Baade:

In principle it is possible to model such a system using two-port networks, however it can get complicated very quickly and it is unlikely in an existing installation that anyone would attempt the modeling. It would be much more logical to measure impedances with an impedance tube that you would stick into the combustion area.

K. McConnell:

Is there any way to shift the frequency up in order to gain stability?

P. Baade:

If you have a way to change the geometry of the combustion chamber, yes. In a practical case for the situation you describe, it is unlikely that you can do anything to much change the natural frequency of the combustion chamber. What you could do is add damping; and if you look at the thing to see where it would do good, you might find a very simple way. What you should not overlook is that damping does not have to be by absorption of acoustic energy. It could be by enhancement of the radiation out of the system into the environment. Now that may sound counter-productive to reducing the generation of noise, but if you radiate it out faster than it can be generated, it won't be generated in the first place!

J. Tichy:

Well, I'm not sure we are talking about damping because if you consider the Nyquist Criterion for feedback and if you can find a way to get it out of phase, then you don't have to add damping.

P. Baade:

That's correct. There are various ways to do it. If you can change the transfer function of the flame, change fuel/air ratio, change burner port dimensions, you can accomplish the same thing without damping.

WOODWORKING MACHINERY NOISE RESEARCH, DEVELOPMENT, AND DEMONSTRATION

F. D. Hart, Professor
J. S. Stewart, Adjunct Assistant Professor
T. H. Hodgson, Associate Professor

Center for Acoustical Studies
North Carolina State University
Raleigh, North Carolina

INTRODUCTION

Interest at the Center for Acoustical Studies (CAS) in noise from woodworking machinery was stimulated by an inquiry from the Newman Machine Company of Greensboro, North Carolina in the fall of 1970. Newman Machine Company had, during the previous year, made efforts to reduce noise of their planers (single and double surfacers) with very disappointing results (even though an acoustical consulting firm had supported their efforts). Experience with planer noise by the CAS personnel at that time consisted of observations made in routine noise surveys in the furniture industry and familiarity with several articles in the literature which referred to aerodynamic noise. Experience at Newman showed that planer noise seemed to be related in some proportion to the following:

1. depth of cut taken on the board;
2. width of board being cut;
3. number of heads doing the cutting;
4. specie of wood being cut;
5. speed at which the material is fed past the cutter.

Specific questions raised by Newman in initial discussions included:

1. what is causing the noise;
2. is it the knife striking the wood;
3. is it the chips being torn from the wood;
4. is it the angle at which the knife strikes the wood;
5. is a shearing cut quieter than a straight cut;
6. does the noise level vary directly with the depth of cut;
7. does the noise level vary directly with the width of cut;
8. does the noise level vary directly with the speed at which you cut;
9. does the noise level vary to any practical degree with the specie being cut?

It became readily apparent that much work would be involved in finding the answers to these questions, but that seeking the answers would serve as a basis for understanding how one might reduce the noise emission. The first decision that had to be made was how to establish the university/industry joint research effort.

Since the CAS personnel knew very little about the design, production, and operational requirements of wood planers and Newman engineers had little experience with noise, a working agreement that tried to optimize what both parties could best contribute to the effort was agreed upon. For the study, Newman Machine, therefore, furnished a planer and a dust collection system, ancillary hardware, special jigs and fixtures, and engineering and technician time while the CAS contributed faculty and graduate students' time, laboratory space, and measurement and analysis equipment and instrumentation.

No direct monetary payments were made initially to the University by the Newman Machine Company. There were several reasons for this. First, the CAS was reluctant to enter into a contractual agreement until sufficient background and experience with a new problem was gained to determine the likelihood of making significant progress towards solution to the problem. Secondly, the CAS had the good fortune during that period of time to be carrying out an acoustics training program under sponsorship of the National Aeronautics and Space Administration. This program did not restrict participants to select research topics related to aircraft noise. As a result, a trainee was given the opportunity to study the wood planer for his research problem. Thirdly, furniture manufacturing and wood products are major industries in North Carolina, and the University encourages faculty participation in problem solving that could have benefit to North Carolina industry.

An initial three month study established vibration of the boards being planed as the mechanism of noise generation. Direct monetary support from Newman Machine Company began in January, 1971. Since that time a number of projects have been conducted. In the following section, summaries are provided of specific project activity; hardware development as a direct or indirect result of the studies is reviewed; a listing of papers and reports and manpower development is provided; and suggestions are given for priorities in future research, development, and demonstration.

WOODWORKING MACHINERY NOISE RD&D PROJECT ACTIVITY

Table 1 gives a summary of projects, support periods, personnel, support sources, and an estimate of total

resources (money, manpower, and equipment) related to woodworking machinery from 1970 to the present. The dollar values given in the far right column do not include the in-house RD&D effort (particularly Newman Machine Company) carried out by the various companies. The 5th entry is included because three of the NASA trainees conducted research on woodworking machinery. An estimate is made in #6 of the value of equipment provided by several companies not accounted for in other listings.

An attempt has been made to estimate the distribution of manpower involved in woodworking machinery noise RD&D over the last six years. This estimate is shown in Fig. 1. It includes part time graduate students and faculty without regard to direct support. It is estimated that between four (4) and five (5) man-years have been devoted to this effort since 1970. Direct outlay of money and equipment provided by industry is estimated at about \$52,000. This figure does not include the engineering time of industry personnel in working with researchers at the CAS on noise problems or the cost of personnel time for meetings and discussions held at various companies.

In project #1, the wood planer was studied. Project #2 required analyzing noise for the wide range of machines used in the lumber, wood products, and furniture industries. Circular saws, high speed routers, and shapers were studied in the laboratory and in the field in project #3. The vibration characteristics and aerodynamic noise of circular saws comprised project #4. Project #5 is included since three NASA trainees did Ph.D. theses on several aspects of woodworking machinery noise. The miscellaneous equipment listing, #6, includes a planer, a high speed router, a shaper, and a circular saw. The proposal for project #7 was submitted in 1972 and funded in 1974. Its objective is to provide a practical guide for planer noise control and to extrapolate those control principles to other woodworking machines.

MAJOR RESULTS AND HARDWARE DEVELOPMENT

In addition to better understanding of the mechanisms of noise generation and more efficient enclosure utilization, the cooperative university/industry R&D effort has led to hardware development that is being used, in part, for noise control purposes on planers, matchers, circular saws for multiple application, tenoners, and molders. A noteworthy example of such hardware development is an helical carbide cutterhead that is being produced by Newman Machine Company under the trade name "Quietcut." Fig. 2 shows this cutterhead installed in an industrial wood planer. This cutterhead gives from 15 to 25 dBA noise reduction as compared to a conventional steel straight knife cutterhead and in many applications reduces operator exposure to less than 90 dBA without the use of enclosures. The design principles and scaling laws upon which this cutterhead was developed are discussed in literature listing #22.

It should be pointed out that using the foregoing example should in no way be interpreted as an endorsement for application. Other semi-helical, segmented, and carbide cutterheads are commercially

available and provide varying degrees of noise reduction.

ADDITIONAL RESEARCH PRODUCTS

An important product that results from university research is the number of students that receive specialized training and graduate degrees (M.S. and Ph.D.). Four M.S. and three Ph.D. students did theses work that relates directly to woodworking machinery noise. Only two of these graduates (John Stewart, Ph.D. and William Feisler, M.S.) are currently engaged in professional activity concerned with these types of noise problems. Dr. Stewart has a business (Noise Control Services, Inc.) that specializes in noise control in the lumber, wood products, and furniture industries.

An effort has been made to present the results of this research in the literature and to make presentations at appropriate professional society meetings. This has resulted in about twelve (12) published papers and eight (8) major reports. Additionally, a number of articles have been prepared for the trade magazines catering to these industries. Several additional papers and reports are in various stages of preparation. A listing of the papers and reports is given as the last section of this paper.

RECOMMENDATIONS

Enough work has been done to raise the level of confusion as to what is technically possible, practical, and economically feasible in control of noise from woodworking machinery. The best way to clear up uncertainties as to the state-of-the-art for noise control in the lumber, wood products, and furniture industries is to develop and implement a major demonstration program in several plants with representative machinery and production operations. A plant would be selected wherein little progress had been made in noise control. A detailed noise analysis would be conducted, and attempts would be made to reduce noise to acceptable levels by the most appropriate combination of tooling changes (resulting from R&D efforts), retrofitting of machines, total and partial enclosures, and other standard noise control procedures.

After all control measures have been implemented, a detailed analysis would be carried out to determine reduction in noise exposure. The operation would be monitored for twelve (12) months following the analysis to determine any undue interference with production, increase or decrease in production cost, and reliability of noise control hardware.

Such a demonstration program would: 1) establish the nominal level to which noise could be controlled, 2) establish the cost of control and the economic feasibility, 3) establish the reliability of noise control hardware, and 4) establish the decrease in incidence of hearing impairment if coupled with an audiometric testing program to run parallel with the engineering noise control demonstration program.

It is recommended that the first next step in woodworking machinery noise RD&D be a majordemonstration

program. The program would establish the state-of-the-art and identify the technology gaps and would, therefore, serve as a basis for realistic and cost beneficial investment of additional RD&D funds.

PAPERS AND REPORTS

1. "Status of Noise Control for Industrial Wood Planers," J. S. Stewart, F. D. Hart, and J. R. Bailey. CAS Internal Report, November 1971, prepared for Newman Machine Company.
2. "Analysis and Control of Industrial Wood Planer Noise," F. D. Hart and J. S. Stewart, Sound and Vibration, pp. 24-27, March 1972.
3. "Analysis of Noise Generation in Industrial Wood Planers," J. S. Stewart and F. D. Hart, presented at ASA meeting, Abstract JASA, Vol. 52, 1972.
4. "A Theoretical and Experimental Study of Wood Planer Noise and Its Control," J. S. Stewart, Doctoral Thesis, North Carolina State University, Raleigh, North Carolina, August 1972.
5. "How to Build Enclosures for Noise Control in Woodworking," J. S. Stewart, Published in MEIHO International Woodworking Survey, Japan, 1973.
6. "Test Code for Evaluating the Noise Emission of Woodworking Machinery," F. D. Hart and J. S. Stewart, Woodworking Machinery Manufacturers of America, 1900 Arch Street, Philadelphia, Pennsylvania, revised, January 1973.
7. "Woodworking Machinery Noise - Sources and Solutions," J. S. Stewart, Woodworking and Furniture Digest, June 1973.
8. "Workpiece Vibration Control in Wood Planers," J. S. Stewart and F. D. Hart, ASME Publication Number 73-DET-79, September 1973.
9. "Investigation of Noise Radiated from a Circular Saw," F. D. Hart, D. M. O'Brien and D. A. Ferioli. Proceedings: NOISE CON 73, pp. 420-422, October 1973.
10. "Noise Control Research Program on Woodworking Machinery," F. D. Hart, J. S. Stewart, et al., Final Report to the Woodworking Machinery Manufacturers of America and the Southern Furniture Manufacturers Association, January 1974.
11. "Noise Reduction Techniques for the Saw Mill Industry," J. S. Stewart, Modern Saw Mill Techniques, Vol. 3, Miller Freeman Publication, 1974.
12. "Noise Abatement Practices," J. S. Stewart, Woodworking and Furniture Digest, November 1974.
13. "Aerodynamic Noise in Idling Woodworking Machines," T. F. Brooks, Doctoral Thesis, North Carolina State University, Raleigh, North Carolina, 1974.
14. "Reduction of Aerodynamic Noise in Idling Woodworking Machines," T. F. Brooks and J. R. Bailey, Proceedings of Inter-Noise 74, pp. 369-372, October 1974.
15. "An Investigation of the Idle Noise of Circular Saw Blades," R. F. Keltie and W. F. Reiter, CAS Report prepared for Gometex Tools, Ltd., Kalmar, Sweden, July 1974.
16. "Mechanism of Aerodynamic Noise Generation in Idling Woodworking Machines," T. F. Brooks and J. R. Bailey, ASME Paper No. 75-DET-47, October 1975.
17. "Dimensional Analysis and Scaling of the Aerodynamic Noise Produced by Idling Circular Saws," R. F. Keltie, CAS Report TR 75-26, August 1975.
18. "On the Nature of Idling Noise of Circular Saw Blades," W. F. Reiter and R. F. Keltie, Journal of Sound and Vibration 44(4), 531-543, 1976.
19. "Dimensional Analysis and Scaling of the Aerodynamic Noise Produced by Idling Circular Saw Blades," R. F. Keltie and W. F. Reiter. Accepted for publication by ASME, December 1976.
20. "The Acoustic Radiation from a Rotating, Resonating Disk," D. M. O'Brien and T. F. Hodgson, CAS Internal Report, In preparation, Fall, 1976.
21. "A Theoretical and Experimental Investigation of a Vibration Suppression Model for an Industrial Wood Planer," M. Black, CAS Internal Report, in preparation, Fall, 1976.
22. "Control of Industrial Wood Planer Noise through Improved Cutterhead Design," J. S. Stewart and F. D. Hart. To be published in the July/August 1976 issue of the Journal of Noise Control Engineering.

Table 1. Summary of Woodworking Machinery Noise RD&D

Project Title	Support Period	Personnel	Support Sources	Amount
1. Wood Planer Noise Analysis	11/1/70-6/15/73	F. D. Hart J. S. Stewart J. R. Bailey	Newman Machine Company	\$10,000
2. Development of Test Code for Noise Emission of Woodworking Machinery	5/15/72-8/31/72	F. D. Hart J. S. Stewart W. K. Feisler	Woodworking Machinery Manufacturers of America (WMMA)	\$8,000
3. Noise Control Research on Woodworking Machinery	10/15/72-8/31/73	F. D. Hart, J. S. Stewart D. M. O'Brien, D. Ferioli, W. F. Reiter, R. Keltie, W. K. Feisler	WMMA and Southern Furniture Manufacturers Association	\$20,000
4. Circular Saw Noise Analysis	8/20/73-6/30/74	W. F. Reiter R. Keltie	GOMEX, Ltd. (Sweden)	\$4,000
5. NASA Acoustics Training Program	9/1/69-8/31/75	F. D. Hart, J. S. Stewart T. Brooks, D. M. O'Brien J. R. Bailey, T. H. Hodgson	NASA	Indirect support - % of \$251,000 total estimated at \$30,000
6. Miscellaneous-Equipment and Tooling Donations	1/1/71-12/31/75	F. D. Hart J. S. Stewart	Newman Machine Co. Nelson Machine Co. Thomasville Furniture Ind. North American Tool Co.	\$10,000
7. Noise Control Research on Wood Planers	4/1/74-8/31/76	F. D. Hart, J. S. Stewart R. Keltie, W. F. Reiter M. Black, D. M. O'Brien, J. Perkins	NIOSH	\$100,000

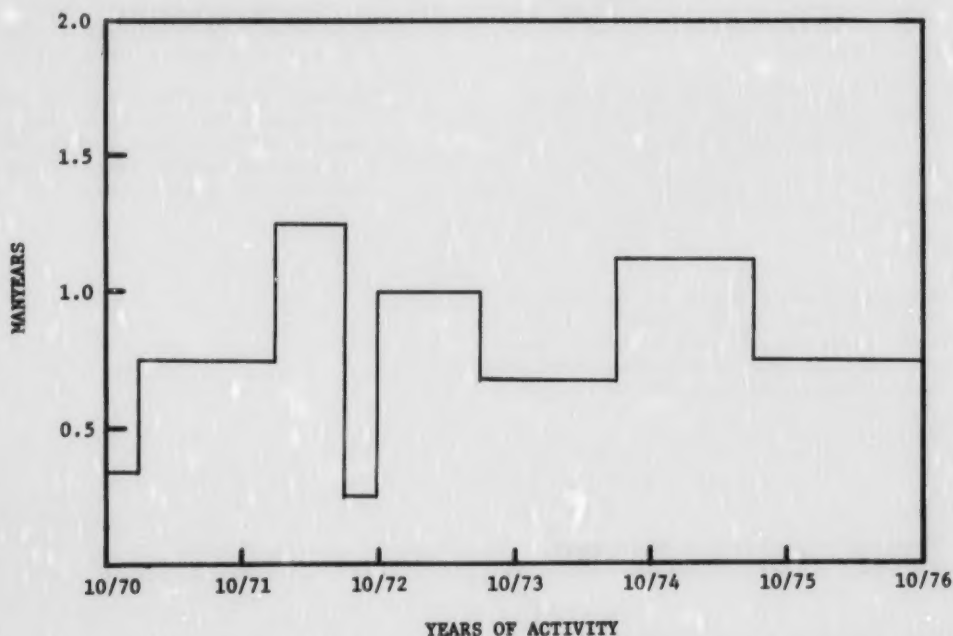


Fig. 1 Estimate of Manpower Committed to Woodworking Machinery RD & D.



Fig. 2 Industrial Wood Planer Equipped with Carbide Helical Cutterheads.

"Woodworking Machinery Noise Research,
Development and Demonstration"
presented by
F. D. Hart

DISCUSSION:

J. D. Gibson:

Frank, I'm not quite clear here, is the enclosure method preferable for wood planers?

F. Hart:

It depends on the companies' monetary situation. We would explain to them that a helical cutterhead would control their noise without the need of enclosures. But the cutterhead costs on the order of 20% of the initial capital cost of the machine--about \$6000 (less installation) for a 30 inch cutting head. But if you can convince them, which is true, that you only have to sharpen it once every three months as opposed to once or twice a day for a standard steel straight knife cutter, maintenance costs are reduced considerably. In fact one economic study done in Alabama with a carbide head showed that in a period of a year you were saving money even discounting the initial capital cost of a very expensive retrofit component.

J. Tichy:

How did you dampen the boards fed into the machine?

F. Hart:

About the damping: in a preliminary study we did use rubber attached by contact cement to dampen one side of a board fed into a single cutting machine. The damping was measurable, but of course this is not a practical solution. The most effective way we found was a distributed contact of steel-on-board, as you would suspect. The problem is the more pressure you put on it, the harder it is to push through.

ELBOW DUCT CROP DRYER FAN EXHAUST MUFFLER

David K. Holger, Assistant Professor
Kenneth G. McConnell, Professor

Department of Engineering Science and Mechanics
and
Engineering Research Institute
Iowa State University
Ames, Iowa 50011

INTRODUCTION AND BACKGROUND

Centrifugal and axial fans are the primary component of many crop aeration and drying systems. Often these fans are the most intense noise sources associated with grain handling operations. In order to reduce the noise generated by such fans, it is necessary to obtain acoustical data which characterize the sound produced by the fans involved. During the summer of 1975 a test program involving 10 axial and 12 centrifugal fans was completed. All of the fans tested are integral parts of either grain drying or aeration systems. The axial fans tested range in size from 12 to 28 inches in diameter and from 1/3 to 10 hp, respectively. Two of the axial fans were two-stage units using double shafted electric motors. The centrifugal fans were all direct drive units with backward curved flat blades. They ranged from 5 to 40 hp and had inlet diameters ranging from 20 to 27 inches.

The objectives of the test program were to obtain baseline acoustic data for each of the 22 fans, to obtain from the baseline data information which would lead to methods for quieting present fans, and to obtain data for use in fan redesign. The experimental program consisted of a series of octave band pressure level measurements at various fan operating static pressures and positions around the inlet and exhaust of each of the fans. For six of the fans complete octave band pressure level measurements were made on ten point hemispherical arrays at two radii. Magnetic tape recordings of the sound generated by each fan were made for subsequent narrow band analysis. The baseline acoustic data thus obtained identified which of the fans were the most serious potential noise problems.

Using the baseline data several fan modifications were attempted with only moderate success as far as noise reduction was concerned. Because of the immediate need for large noise reductions at several installations, it was decided to proceed with muffler design for two of the centrifugal fans which had been tested. In this case the immediate need had priority over the actual sound output of the fans. This paper presents the results of some laboratory and field tests conducted for the two centrifugal fans of interest. Pressure level mea-

surements are presented for both fans with and without the elbow duct mufflers which were designed to provide the necessary reduction in pressure levels. With few restrictions the design should be applicable to the inlet or exhaust sides of either axial or centrifugal fans of small to moderate size.

THE MUFFLER

The muffler which was designed for use on the exhaust side of centrifugal fans with 20 inch and 27 inch diameter inlets is sketched in Fig. 1. It consists of a sheet metal housing with two internal baffles in the shape of a 90 degree elbow. To meet geometric constraints at various field locations the elbow angle was reduced to as low as 30 degrees in some cases. For smaller angles the duct was lengthened downstream of the elbow so that there was no unobstructed line of sight through the muffler. The interior surfaces of the housing and flow side of the baffles were lined with 3 inch thick fiberglass blanket covered on the exposed side with 1/2 inch hardware cloth. In order to minimize the back pressure produced by the muffler, it was designed so that the area of the open cross section of the muffler increased slightly from the fan exhaust to the open end. All measurements indicate that the muffler had a negligible effect on fan performance.

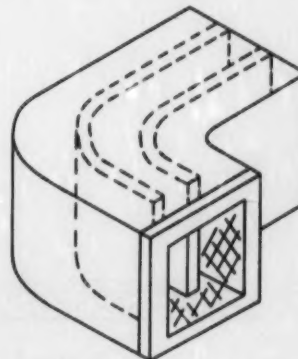


Figure 1. Elbow duct muffler.

At least two problems could arise in the use of an elbow duct muffler in other types of fan applications. First there is the size and geometry problem. For large fans or fans in a size restricted space, the elbow duct may be too large for installation. All mufflers constructed as part of this work were slightly larger than the fans upon which they were mounted. This bulkiness is primarily a result of the need to have a long enough elbow to block line of sight through the muffler. For large fans the size also means that the weight of mufflers is large enough so that more support is needed than is provided by simply bolting the muffler to the fan exhaust. A second potential disadvantage is that for high flow rates there is an erosion problem with the fiberglass lining. This means that a more expensive foam type absorbent material must be used for the duct lining. For the short term tests conducted, maximum flow rates in the vicinity of 10,000 cfm caused no significant erosion problems. For the fans tested this flow rate would cause maximum air velocities of around 40 to 50 ft/sec.

LABORATORY TEST

Laboratory tests were conducted on a 27 inch 10 hp centrifugal fan which was capable of delivering from 2000 to over 10,000 cfm at 8 and 2 inches of water, respectively. The fan had eight backward curved flat blades and operated at 1745 rpm. The test area consisted of a roughly circular 25 foot diameter enclosure formed using 4 foot by 8 foot flats of 6 inch thick fiberglass on 3/4 inch plywood backing. A sketch of a typical experimental setup is shown in Fig. 2. The fan was attached to a standard AMCA 22 inch test tunnel, and the static and dynamic pressure were monitored throughout a test run using pitotstatic tubes. All acoustic measurements were made using a General Radio 1933 (Type 1) precision sound level meter with a one inch microphone. The microphone was remotely mounted on a tripod and attached to the SLM by extension cables during a test. The instrument was calibrated at least twice daily using a General Radio 1562-A sound level calibrator. Octave band pressure levels were measured at an array of posi-

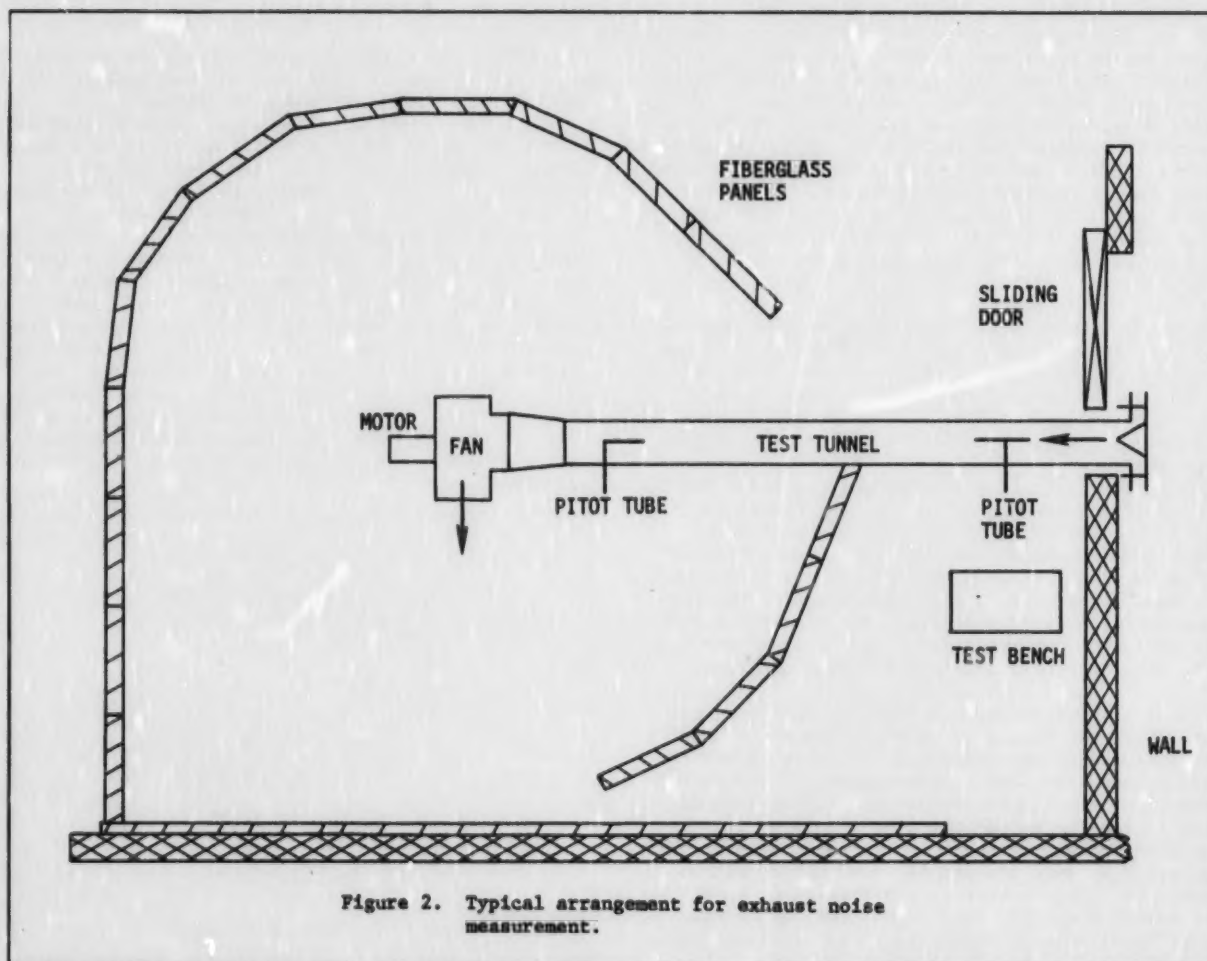


Figure 2. Typical arrangement for exhaust noise measurement.

tions around the fan exhaust, and for a position on the exhaust axis measurements were made for six static pressures.

Figure 3 is a comparison of overall sound pressure level at eight measurement positions with and without the muffler. It should be noted that without the muffler the fan was exhausting horizontally, and with the muffler in place air was exhausted vertically upward. Measurement positions 2-8 are all 8 ft from the fan hub, and position 1 is 5 ft from the hub on the side away from the exhaust. Position 5 is centered in the exhaust for the unmuffled case. The large reduction in level at position 5 is, thus, partially due to absorption by the muffler and partially due to a redirection of the sound upwards. In most grain handling and storage operations, it is only the total reduction in level at sensitive receiver locations which is important. Thus, appropriate redirection of sound is often as good as absorption.

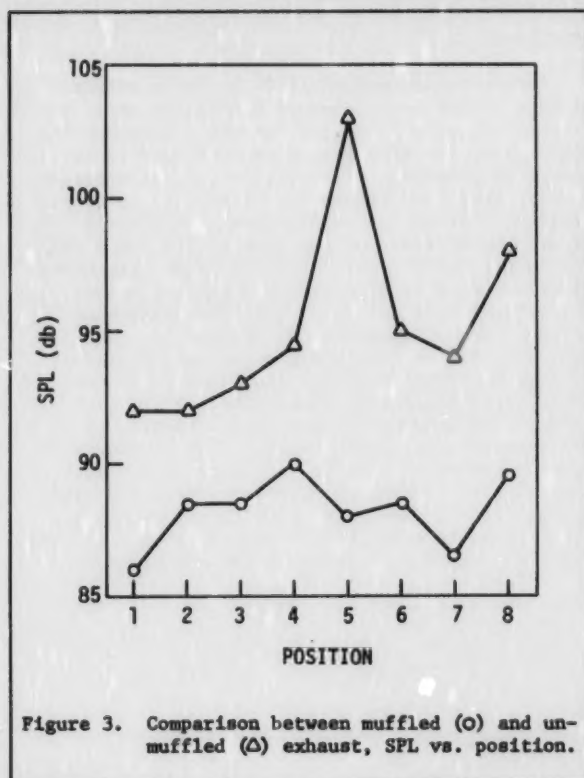


Figure 3. Comparison between muffled (O) and unmuffled (Δ) exhaust, SPL vs. position.

Figure 4 contains a comparison of octave band pressure levels at position 5 with and without the muffler. The data at 31.5 and 63 Hz are somewhat questionable because of test facility limitations. However, reductions appear to be significant at all frequencies. It is speculated that further reductions could be achieved in the lower octave bands through the use of vibration damping materials on the fan and muffler housings. Measurements at various static pressures indicate that the results shown in Figs. 3 and 4 are achieved for the entire

operating range of the fan.

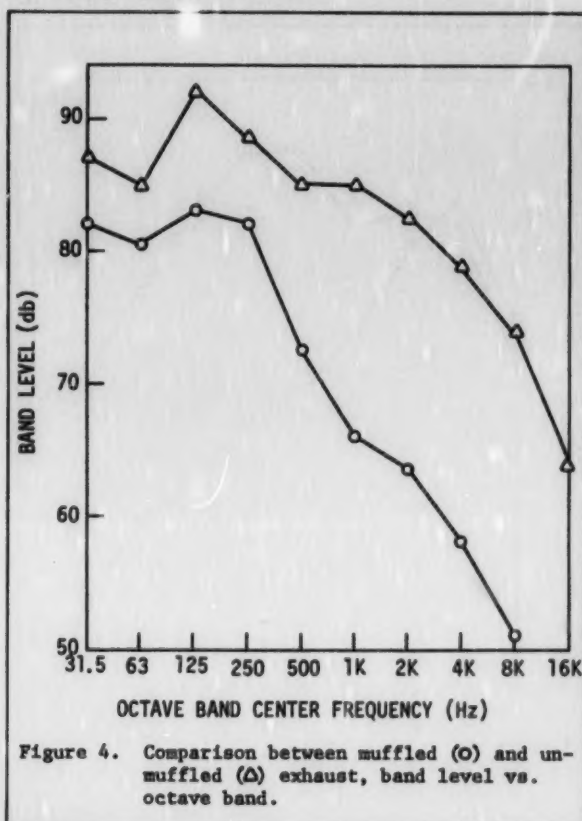


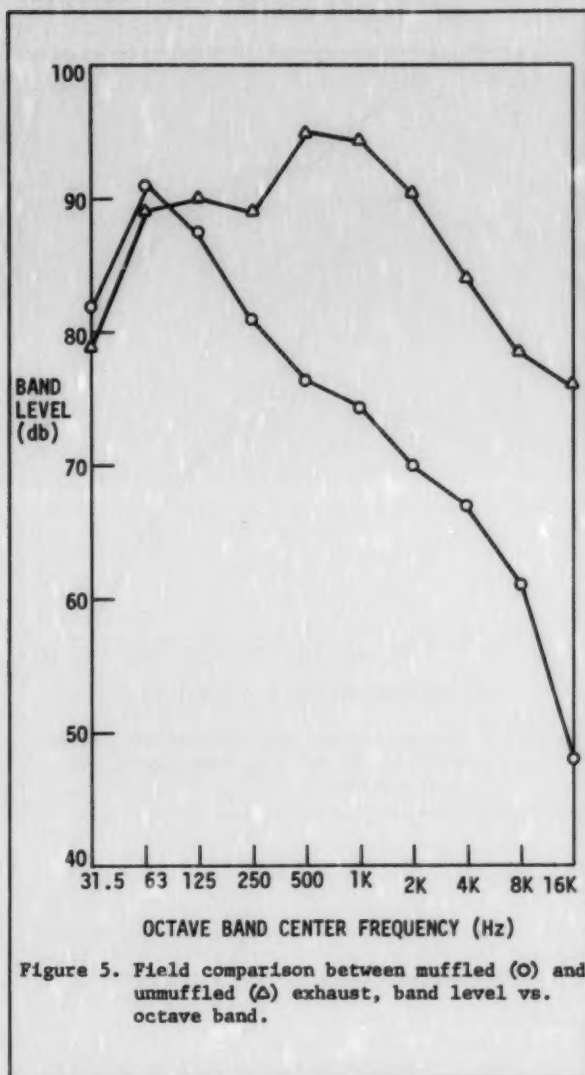
Figure 4. Comparison between muffled (O) and unmuffled (Δ) exhaust, band level vs. octave band.

FIELD TEST

Field tests were conducted on a 20 inch 20 hp centrifugal fan capable of delivering from 3000 to over 10,000 cfm at over 16 and 2 inches of water, respectively. The bin to which the fan was attached was nearly empty so that the fan was operating near the low static pressure -- high flow rate of its range. Measurements were again made with the General Radio 1933 SLM at various distances from the fan exit. The two days on which the measurements were made were both clear days with little wind and temperatures of approximately 65 degrees F.

Measurements at a distance of 20 feet from the exhaust of the muffled and unmuffled fan are compared in Fig. 5. In the two lowest octave bands the muffler appears to slightly increase the pressure level. This is probably the result of panel vibrations of the muffler or transition section from the fan to the muffler for this particular case. Stiffening of the large panels together with application of a suitable damping material to the muffler housing would probably eliminate the sound enhancement at low frequencies. Above 125 Hz the muffler is quite effective in reducing pressure levels in all octave bands. In fact, all bands above 250 Hz have pressure level reductions of more

than 15 dB.



As was the case with the laboratory tests, these reductions represent the total reduction at a point of interest. The reductions are partially due to absorption by the muffler but also contain a contribution from the redirection of the exhaust. At the field installation tested, the fan without the muffler exhausted horizontally in the direction of a private dwelling. The muffler directed the exhaust down toward grass covered ground. The measurements of Fig. 5 were made at 20 ft from the unmuffled exhaust on the centerline of that exhaust. Thus, Fig. 5 provides a reasonable estimate of the reduction in levels experienced at the private home which is a particularly sensitive point in this situation.

SCOPE OF THE EFFORT

The work reported in this paper involved approximately five man days. Included in the five days are two of data acquisition, one of data analysis, and two of muffler design. Measurements were made in the lab for inlet and exhaust sides of both fans. Measurements with the muffler in place were made in the lab for one fan and in the field for the other. Field measurements of the unmuffled exhaust of the 20 inch fan were also made. The data analysis included narrow band frequency analysis of both the fans with and without the mufflers. Data acquisition and analysis time were significantly reduced because the work reported is a portion of a larger test program involving 22 fans. Once the elbow duct shape had been chosen, the design of the muffler was reduced to specification of materials and dimensions. Dimensioning was critical as the line of sight through the muffler had to be completely blocked while keeping the muffler size within constraints of the field location.

CONCLUSIONS

The muffler design presented is a simple effective solution to the noise problem created by small to intermediate size fans used for grain aeration and crop drying. In many situations the muffler is capable of providing the level reductions necessary to bring fans into compliance with state noise regulations. The design should work on either the inlet or exhaust sides of any type of fan. The muffler is particularly effective if level reductions can be enhanced by directing the muffler exhaust in an inoffensive direction or toward an absorbent surface. One disadvantage of the muffler is its size, but this is rarely a problem in grain handling operations. A second disadvantage is that muffler cost increases significantly if special absorbent lining materials are needed because of high flow velocities or temperatures. The mufflers tested appeared to have no influence on fan performance. Thus, in many applications the lined elbow duct muffler is capable of providing convenient effective noise reduction for small to intermediate size fans with no sacrifice in fan performance.

"Elbow Duct Crop Dryer Fan Exhaust Muffler"
presented by
D. K. Holger

DISCUSSION:

D. Holger:

We put in about one man year on this work with funding on the order of ten to fifteen thousand dollars. Actually, a good deal of time was donated.

D. Muster:

In Figures 3 and 4, how can you get 15 dB Insertion Loss at point 5 in Figure 3 when in Figure 4 if you add up the bands you get nowhere near that? Also are the levels flat or A-weighted?

D. Holger:

The levels are flat. It's not mentioned in the Figures, but the data in Figure 4 was obtained under different operating conditions than that in Figure 3. Figures 3 and 4 are internally consistent as far as Insertion Loss is concerned.

P. Baade:

In Figure 3, it seems to me that the amount by which position 5 exceeds the other positions is a little more than that for which I would expect directivity to account. Looking at Figure 4, and bearing in mind the difference in operating conditions, it seems most likely the greatest contributions come from the low frequency bands around 125 Hz. With a two foot opening you certainly would not get the kind of directivity Figure 3 implies. Could there have been flow interaction noise on the microphone for that position?

D. Holger:

We did have wind screens on the microphones, but of course it is a possibility. I might add that on the inlet side where flow is not a factor, we did notice directivity, but not quite that much.

RESEARCH INTO ABATEMENT OF AIR-ABRASIVE BLAST NOISE

John E. Sneckenberger, Associate Professor

Mechanical Engineering and Mechanics
West Virginia University
Morgantown, West Virginia 26506

ABSTRACT

The objective of the research reported in this paper was to reduce the noise level produced by the air-abrasive discharge through the nozzle of an industrial blast system without appreciably affecting the existing surface improvement performance. Experimental development using noise reduction techniques extrapolated from aerodynamic jet noise abatement methods led to several improved nozzle and add-on silencer designs. The experimental facilities developed for measurement of sound pressure level, abrasive flow rate and abrasive velocity are described.

INTRODUCTION

Blast systems which provide mechanical improvement to surfaces, such as sand cleaning and shop peening, have long been designed to satisfy a wide range of industrial requirements. Some such blast systems, however, are now identified as major sources of noise. Thus, in addition to providing efficient performance for surface improvement, such blast systems will need to accomplish their function at reduced noise levels.

PROBLEM DESCRIPTION

The blast nozzle for which improved nozzle and add-on silencer designs were developed during this investigation is shown in Fig. 1. This nozzle is widely used in suction feed blast cleaning cabinets and

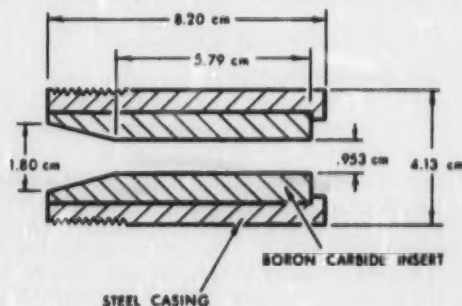


Fig. 1 Existing blast nozzle design

rooms as well as direct pressure blast tank operations. Typical operating nozzle air pressures are 41.4 N/cm² and 55.2 N/cm². These pressures produce a highly turbulent supersonic flow of air-abrasive mixture through the blast nozzle. The resulting sound pressure level, when measured at the location shown in Fig. 2 for air-abrasive blasting inside a cabinet without a workpiece, was found to be 96 dBA at 55.2 N/cm². Other experimentation has indicated that without the cabinet the continuous noise level measured at the same location could be as high as 103 dBA.

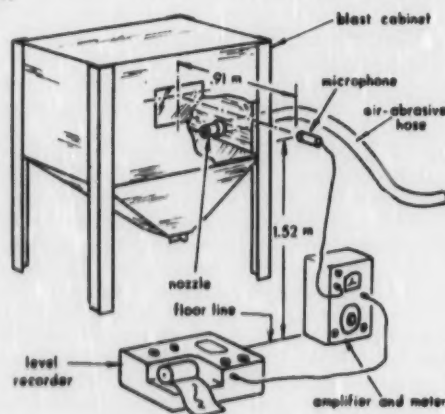


Fig. 2 Arrangement for sound pressure level measurement

Efficient surface improvement performance using this nozzle is largely a function of such parameters as abrasive flow rate and abrasive velocity. These performance parameters were measured during this project. Other performance parameters would be blast pattern and silencer wear resistance, but the measurement of neither parameter was included in the scope of this investigation.

EXPERIMENTAL FACILITIES

A schematic diagram of the experimental facilities developed for this research is shown in Fig. 3. The blast system used during this research project included, in addition to the blast nozzle with/without silencer, an air compressor and filter, an air supply

tank, an abrasive tank, a blast cabinet, and the appropriate connection hoses and regulators. The air compressor was used to fill the air supply tank before each test. The air supply tank was connected

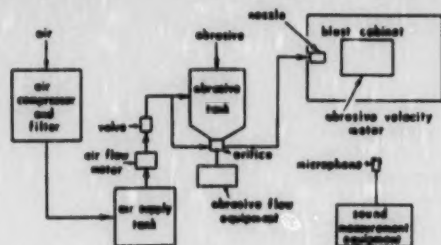


Fig. 3 Schematic diagram of experimental facility

through an air regulator to the abrasive tank. This compressed air forced the air-abrasive mixture from the abrasive tank through a high-pressure hose to the blast nozzle connected into the blast cabinet. An abrasive velocity meter was placed in the blast cabinet and, for matters of convenience, remained there in operation or shielded from air-abrasive blasting during all experimentation.

MEASUREMENT OF PERFORMANCE PARAMETERS

For evaluating the various nozzle or silencer designs, the particular performance parameters of interest were the overall A-weighted sound pressure level, the abrasive flow rate, and the abrasive velocity. Measurement systems for each of these parameters are discussed in the following paragraphs.

The sound pressure level measurement system utilized a 1.27 cm microphone mounted at a location 1.52 m above the floor and 0.91 m from the front vertical centerline of the blast cabinet as shown in Fig. 2. This microphone was connected to a microphone amplifier and meter and the output recorded on a level recorder. Sound pressure levels were measured with the abrasive velocity meter inside the blast cabinet and pushed to the rear of the blast cabinet away from the blast nozzle. A rubber mat was placed over the front of the velocity meter to prevent abrasive from striking the meter and generating impact noise. The laboratory air compressor had an overall sound pressure level of 86 dBA measured at the microphone location. Thus, the air compressor was turned off when noise level measurements were being made. Overall sound pressure level measurements in dBA were typically made for nozzle pressures of 41.4 and 55.2 N/cm² using steel shot abrasive.

The measurement system to determine abrasive flow rate used a circular load ring as a weight-sensing transducer. The abrasive tank was suspended above the floor with the load ring as a series element in the hoist system. This arrangement permitted any variation in abrasive weight to be detected by the circular load ring. Strain gage lead wires from the load ring were connected to a carrier amplifier. The output from the amplifier was then connected to the y-t recorder to provide a permanent time history of the weight of abrasive being discharged through

the nozzle. From these time histories, instantaneous abrasive flow rates could be determined. Abrasive flow rate measurements were typically made at 41.4 and 55.2 N/cm² nozzle pressure for steel shot abrasive. The measurement system is shown perspective in Fig. 4.

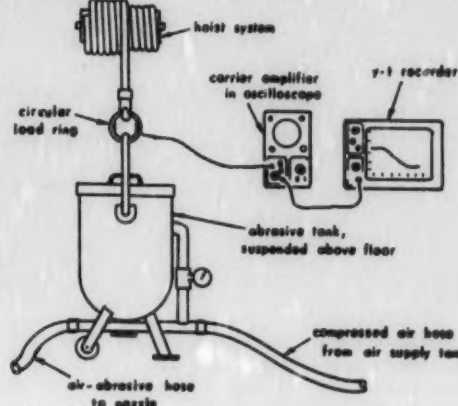


Fig. 4 Abrasive flow rate measurement system

The velocity of the abrasive was obtained by experimentally determining the amount of time necessary for the abrasive to travel a fixed distance. This was accomplished with the use of the abrasive velocity meter shown diagrammatically in Fig. 5. In this abrasive velocity measurement device, two 45.8 cm metal disks are mounted 22.9 cm apart on an axle driven by an electric motor. A 1.27 cm hole was drilled inside the outer edge of the first disk,

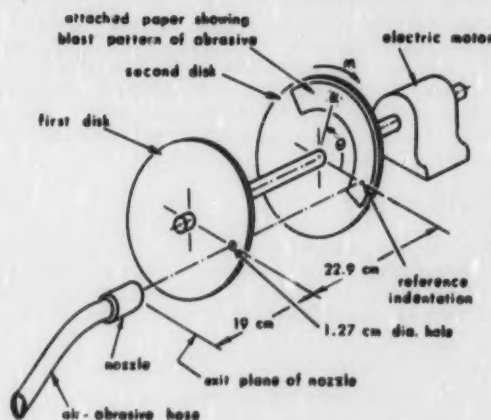


Fig. 5 Abrasive velocity measurement system

and a corresponding reference indentation was made on the second disk in line with the hole in the first disk. When the bore of the blast nozzle lined up with the hole in the first disk, the abrasive would pass through this hole and strike the moving second disk at some angle away from the reference indentation. By measuring this angle and knowing the angular speed of the disks, it was possible to determine the time taken for the abrasive

to travel 22.9 cm. These two quantities of time and distance could then be converted to give abrasive velocity.

DESIGN APPROACH

Initial evaluation of the blast nozzle noise suggested that for the existing blast nozzle, the sound pressure level is largely a function of the air flow rate and relatively insensitive to whether abrasive is or is not being discharged. Hence, techniques which directly attempt to reduce aerodynamic noise appear to be most promising for reducing the blast nozzle noise. Recent literature contains several innovative concepts in aerodynamic noise abatement.

These noise abatement techniques investigated by various researchers can be grouped into the following three main categories;¹

- (a) Use of flow interaction: A secondary flow is used to interact with the primary flow.
- (b) Use of ejector shroud: An entrainment flow mixes with the main flow.
- (c) Use of external mechanical modification at the nozzle exit: Teethed-exit nozzles, center plug nozzles and multitubed nozzle arrangements are examples of such techniques.

The design approach adopted for this project specified that principally because the existing blast nozzle bore dimensions were well established in terms of ease of manufacture and effective surface improvement performance, noise reduction techniques pursued would principally be of the form of blast nozzle exit flow improvements. Any such techniques which showed merit could later be considered for incorporation into the existing blast nozzle design.

The important criteria applicable to the blast nozzle design, in addition to the acoustic criteria which specified the desired noise reduction and the performance criteria which specified that the existing blast nozzle efficiency be retained, included the geometric criteria that specified the restriction of physical dimensions and accessories, the mechanical criteria that specified easy manufacture using durable or replaceable parts, and the economic criteria that considered initial and operating costs.

IMPROVED NOZZLE DESIGN

In the development of the following improved blast nozzle designs, attempts were generally made to incorporate existing aerodynamic noise reduction design features which would reduce the mean shear and turbulence produced by the mixing of the air-abrasive flow from the blast nozzle with ambient air. After each improved blast nozzle configuration was initially fabricated from preliminary design calculations and experimentation begun, further improvements to these configurations were then obtained by altering the various configuration parameters in order to experimentally obtain the maximum noise reduction. The three most attractive improved nozzle designs which were developed are briefly described in the following paragraphs.

The coaxial flow blast nozzle design attempts to use secondary air discharged coaxially around the

periphery of the air-abrasive flow to eliminate or weaken shock related acoustic sources, to enhance the turbulent mixing process, and to reduce the length of the potential core region. This type of blast nozzle design has been reported to reduce supersonic flow noise by about 10 dBA. An experimental blast nozzle model represented by the drawing shown in Fig. 6 was developed. Air hose fittings on the casing allowed secondary air at the

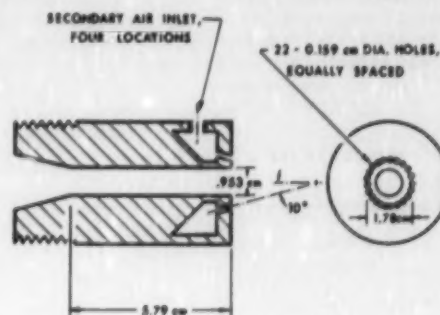


Fig. 6 Coaxial flow blast nozzle design

same pressure as the blast nozzle air pressure to enter the model and to surround the air-abrasive flow. To study the effect of impingement angle of the secondary airflow on the air-abrasive flow, three different end plates with peripheral holes at angles of 0, 10, and 30 degrees to the air-abrasive flow were machined and evaluated.

The ejector shroud technique was developed for both a forced entrainment flow and a self-entrained flow. In addition to the potential noise reduction resulting from the entrainment flow mixing with the air-abrasive flow inside the shroud to produce a uniform flow at the exit of the ejector, the ejector shroud also provides a favorable shield of the aerodynamic acoustic sources. Passive entrainment flow is usually not as effective a noise suppression technique as controlled secondary flow interaction with the main flow. A representative drawing of the developed forced ejector blast nozzle design is shown in Fig. 7.

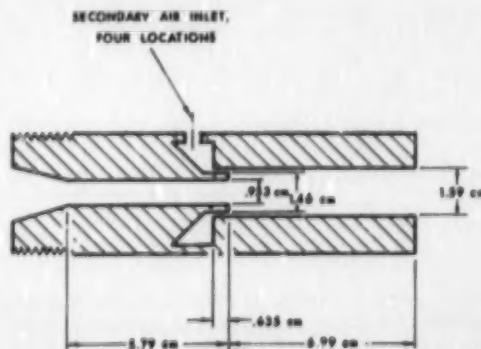


Fig. 7 Forced ejector blast nozzle design

External flow modification techniques at the nozzle exit were employed in the development of the expanded barrel blast nozzle design. The expanded barrel blast nozzle design allows an enlarged annular space in which the air-abrasive flow can expand somewhat to diffuse and dissipate a part of its energy. Many variations of barrel diameters, barrel lengths, and number, size and location of barrel holes were experimentally evaluated. The most favorable blast nozzle design experimentally developed in terms of the design criteria established is shown by representative drawing in Fig. 8 for the expanded barrel blast nozzle.

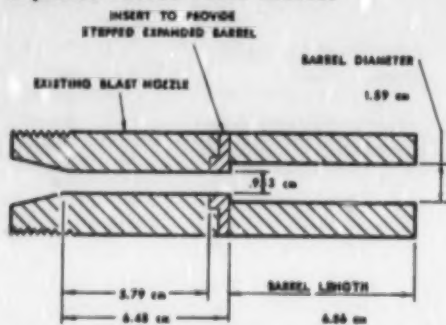


Fig. 8 Expanded barrel blast nozzle designs

Each configuration of a particular blast nozzle design was operated at a number of nozzle air pressures and, at each pressure, the various performance parameters were measured. A summary of the performance results for the existing blast nozzle and three more promising improved blast nozzle designs are given in Table 1 for a nozzle pressure of 55.2 N/cm² using steel shot abrasive. The values shown for abrasive flow rate and abrasive velocity were obtained from one measurement for each blast nozzle. The improved blast nozzle design test program was thus evaluative rather than exhaustive.

Table 1 Experimental Results

Type of Blast Nozzle	Abrasive Flow Rate	Abrasive Velocity	Sound Pressure Level
	kg/min, ±2 kg/min	m/sec, ±2 m/sec	dBA, ±5 dBA
Existing Nozzle	14.7	65.8	96
Coaxial Flow	15.4	69.5	94
Forced Injector	15.1	67.4	90
Expanded Barrel	15.0	60.4	87

* Nozzle air pressure = 55.2 N/cm²

The attractive acoustic results for the expanded barrel blast nozzle were further analyzed. An octave band analysis of the noise from this blast nozzle noise as well as the existing blast nozzle when measured at the microphone location shown in Fig. 2 produced the spectrum analysis results shown in Fig. 9.

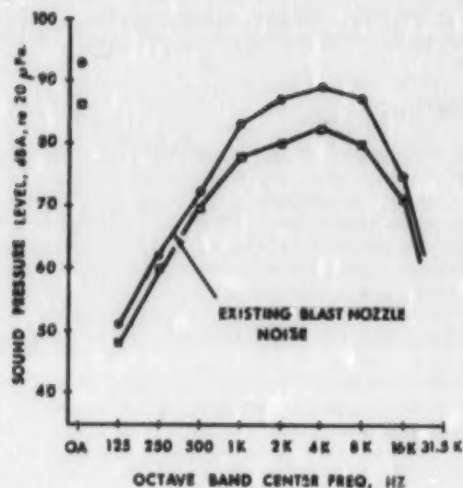


Fig. 9 Spectrum analysis for expanded barrel blast nozzle noise

ADD-ON SILENCER DESIGN

In the development of the experimental silencer designs, sound attenuation at specific frequencies was theoretically attempted where possible. The dimensions of the silencer designs were generally ascertained by starting with certain available parts and then varying certain silencer parameters, largely by trial and error, to provide maximum noise reduction. The following paragraphs provide a description for each of three particular silencer designs that were developed. As mentioned previously, the theoretical considerations described below were utilized only for the purpose of initiating possible experimental silencer designs. Some theoretical considerations proved rewarding while others were not successful.

For developing a Helmholtz resonator silencer for the nozzle flow, helpful design curves were available for predicting values of noise reduction. Using these curves and design formula, a Helmholtz silencer design was developed for a fundamental effective frequency of about 2,000 Hz as well as higher harmonics. To reduce the expected wear problem resulting from abrasive flow, a flared Helmholtz silencer was designed as shown in Fig. 10.

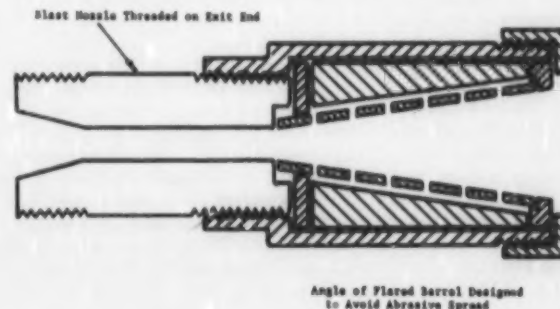


Fig. 10 Helmholtz resonator silencer: flared barrel

A simple expanded volume silencer consists of an abrupt change in the cross-sectional area of a pipe, resulting in an enlarged barrel volume. Theoretically, at certain frequencies, the expanded volume will provide a mismatch of acoustic impedance for the nozzle flow at the enlarged volume interfaces. Such impedance changes will cause incident sound waves to be partially reflected back and interfere with other incident waves, resulting in noise reduction. As a first design of an expanded volume silencer, a previously conceived and developed silencer was experimentally adjusted to provide maximum noise reduction at about 6,000 Hz. This was the lowest effective frequency that was conveniently practical with this available silencer. A descriptive drawing of this expanded volume silencer is shown in Fig. 11.

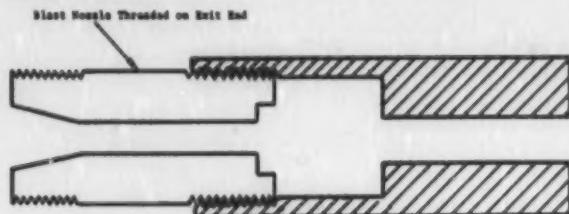


Fig. 11 Expanded volume silencer

In the enlarged barrel silencer design, the nozzle noise conceptually will be reduced by allowing the flow from the nozzle to partially expand in an additional length of larger diameter barrel. This enlarged barrel allows the flow to be subsequently slowed down before being discharged into the ambient air. The mixing process between the flow leaving the enlarged barrel and the ambient air should be less turbulent than that mixing process for the unsilenced nozzle. Hence, some noise reduction should result. As shown in Fig. 12, an enlarged barrel insert was fitted into a steel casing and the casing threaded onto the end of the blast nozzle. Several enlarged barrel silencers were experimentally developed with different lengths and diameters of enlarged barrel to provide alternative optimum surface improvement performances.

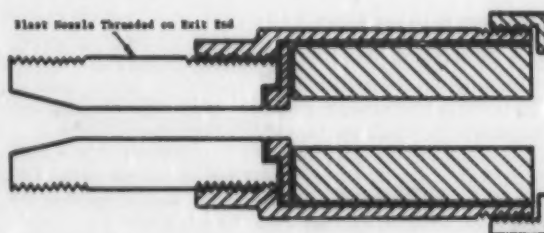


Fig. 12 Enlarged barrel silencer

Experimental results for these three add-on silencer designs developed are summarized in Table 2 for a nozzle pressure of 55.2 N/cm² using steel shot abrasive. These results were obtained from one test of each design.

Table 2 Experimental Results

Blast Nozzle with	Abrasive Flow Rate	Abrasive Velocity	Sound Pressure Level
	kg/min, ±.2 kg/min	m/sec, ± 2 m/sec	dBA, ±.5 dBA
No Silencer	14.7	65.8	96
Flared Helmholtz	14.4	57.9	96
Expanded Volume	14.6	66.4	95
Enlarged Barrel	15.0	60.4	87

* Nozzle air pressure = 55.2 N/cm²

CONCLUSIONS

It has been shown that experimental development using noise reduction techniques extrapolated from turbulent flow theories and abatement practices has led to several improved blast nozzle and silencer designs. Furthermore, the noise reductions have been achieved for most improved blast nozzle and silencer designs with no appreciable reduction in terms of surface improvement efficiency as indicated by abrasive flow rate and abrasive velocity.

Several other conclusions can be drawn from the results of the improved nozzle design test program. Observations made during the tests and the data indicate that the improved blast nozzle designs effect noise reduction in several ways. The flow of secondary air, as with the coaxial flow nozzle, provides some noise reduction. However, the more significant noise reduction resulted simply from an enlarged annular barrel extension. The diameter and length of the barrel extension were more important to the noise reduction achieved than were various openings and entrainment flows.

The Helmholtz resonator design was the first silencer concept developed. The initial evaluations, performed with air only for a straight barrel design, were quite encouraging and suggested that the Helmholtz principle may provide reduced noise levels for the nozzle noise. However, further experimental development of this silencer design, in which the number and size of the tubular openings connecting the branched volume were varied, indicated that the large part of the noise reduction achieved resulted from the enlarged diameter of the added barrel. Thus, it was later concluded that the Helmholtz principle itself does not make the significant noise reduction contribution in this silencer design. Although the flared Helmholtz resonator silencer did not produce attractive test results, they were included because of the resonator's minimum wear characteristics.

The expanded volume silencer concept as experimentally developed during this project did not lead to any noticeable noise reduction. Absorptive material filter assemblies were fabricated to fit into the expanded volume portion of the silencer, but no improvement in noise reduction was observed.

The enlarged barrel silencer provided the best overall solution to blast nozzle noise reduction in terms of the various design criteria specified. Alternative proportioning in enlarged barrel dimensions during experimental development enabled optimum values to be obtained for several of the performance parameters. Table 2 shows that a noise reduction of 9 dBA was achieved.

Recently, work has been done by some researchers to obtain computer-aided silencer designs. Such computer-aided designs could save a lot of time and hardware work in trying various silencer concepts and configurations. Results from such designs and further experimentation should lead to generalized relations for the design parameters of effective silencers such as the enlarged diameter silencer.

It should be remembered that these conclusions which are for air-abrasive flow through the blast nozzles and silencers can be expected to be different from the conclusions that would result if air alone were the flow through the nozzle. See Appendix. In this study, for example, the effect of abrasive in the blast nozzle flow was found to be a decrease in noise reduction achieved by a modified blast nozzle design. Hence, a new blast nozzle or an add-on silencer that reduces airflow noise may not reduce air-abrasive flow noise, even for the same airflow rates.

ACKNOWLEDGEMENTS

The project to develop a silencer for the blast nozzle was supported by the Pangborn Division of the Carborundum Company, Hagerstown, Maryland. The Pangborn Division also provided the blast system and numerous other forms of assistance, such as the abrasive velocity meter, for the experimentation. The contributions of John Pellegrin and Kalyan Bagga as Graduate Research Assistants is also gratefully acknowledged.

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APPENDIX: Air Only Discharge Through Improved Nozzle

The most significant noise attenuation for exhausted compressed air with no abrasive was achieved using the segmented expanded barrel diffuser type of nozzle modification shown in Fig. 13. This step change in barrel cross section permits the flow to be partially expanded and subsequently slowed down before being discharged into the ambient air. Dimensions of the entire nozzle, particularly those of the segmented barrel, are critical to the effective performance of the expanded barrel nozzle.

Studies were performed to determine the effective performance range for several of the barrel diffuser parameters, such as d_p in Fig. 13, in terms of various flow parameters as well as the resulting noise level. The effect of various

expanded barrel diameters on measured sound pressure level as a function of nozzle pressure is shown graphically in Fig. 14. Comparison of these curves for different diameters with the 'nozzle only' curve indicates that a 15 dB noise reduction can be attained for at least a 7 N/cm² range about operating nozzle pressures of 25, 40 and 55 N/cm². As could be expected, other intermediate size diameters would provide similar nozzle noise reduction for other values of operating nozzle pressures.

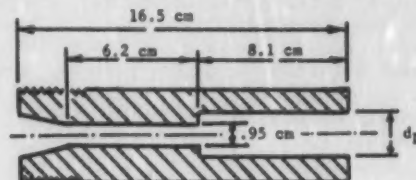


Fig. 13 Expanded barrel diffuser

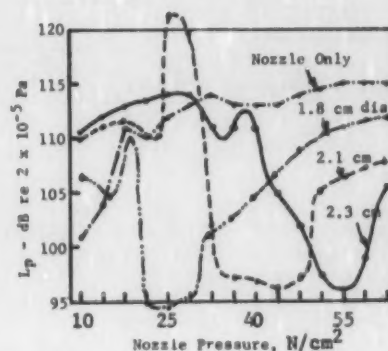


Fig. 14 Sound pressure levels for various expanded diameters

In summary, thus, it has been experimentally found that barrel diffuser parameters for the expanded barrel nozzle can be selected to provide a reduction of as much as 20 dB for air flow noise generated at typical nozzle operating conditions. Additional studies of this noise reduction technique are currently being contemplated.

The simplicity of the developed flow noise reduction technique invites application to silencing other similar noises such as turbulent pipe flow noise, cleaning nozzle noise and possibly engine exhaust noise. Many chamber and blowoff silencer designs for gases involving flow baffling and porous material dispersion are often unsuited for applications in which the gases contain a particulate. The expanded barrel technique is quite suited to this type of noise reduction application.

"Research into Abatement of Air-Abrasive Blast Noise"
presented by
J. E. Sneckenberger

DISCUSSION:

D. Muster:

First of all I want to compliment you. This is a good paper in that you've got the SI System and you've got your data presented so that you can read it. But the thing that concerns me is the velocity device. It's not an averaging device as I see it. I've made measurements with very thin wires and air damping can be tremendous, especially when you get down to small particulates, so that what you see is the lowest value, not an average.

J. Sneckenberger:

My definition of an average velocity is this: the particle travels through the first hole, then in the process of traveling it decreases its velocity over the distance between the two disks. Since we always kept the disk spacing constant, the measurement served as a comparison between different designs.

D. Muster:

What was the air velocity?

J. Sneckenberger:

I don't know. That would require a hot wire anemometer, and with the abrasive in there, well.....

K. McConnell:

Have you run with pressures higher than what you reported in the paper?

J. Sneckenberger:

Eighty was the highest we had reported, and to go to higher pressures would have required additional compressor facilities that we did not have at the University.

R. Cohen:

I guess the expanded barrel is the best from your studies. How about a variation of parameter study--geometric or otherwise--to optimize for a particular application? Have you done this?

J. Sneckenberger:

I have another slide here for air only. Here we have the expanded barrel configuration discharging air only. This is the graph for different diameters showing sound pressure level versus nozzle pressure. What happens when we change the expanded diameter port from 11/16, 13/16, 14/16-inch is that the pressure for which maximum attenuation is achieved moves upward. We also tried varying the length, but that does not appear to be too important, although I've seen some recent work from Poland showing that the length is important.

R. Lyon:

What part of this project was done by your engineering acoustics class and what part was done by subsequent thesis, and would you comment on the general educational impact of the project?

J. Sneckenberger:

The engineering acoustics class got the equipment in at about mid-semester and their result was the graph that showed the sound pressure level with and without abrasive. With that result the company wanted us to continue with more work. Thus the engineering acoustic class in one-half a semester got the equipment going and showed that result. I think they benefitted in that it took them from the textbook to a real industrial problem. As far as graduate study, we had two graduate students in the extended work. Both students had a chance to become familiar with some of the real up-to-date ideas in noise control, and also they had to have the hardware built, etc. Both of them seemed to have been very satisfied with their efforts.

NOISE AND VIBRATION STUDY OF SMALL HERMETIC REFRIGERANT COMPRESSORS

James F. Hamilton
Professor of Mechanical Engineering
Ray W. Herrick Laboratories
Purdue University
West Lafayette, Indiana 47907

INTRODUCTION

Increased consumer reaction to household appliance noise levels is producing noise reduction projects by many appliance manufacturers. Low noise levels have become a desirable sales feature. Because of the intermittent operation of freezers, refrigerators, and air conditioners, consumers are more sensitive to the noise levels of these appliances than to the noise levels of many other appliances.

Since the refrigerant compressor is one of the major sources of noise in these appliances (1) a general knowledge of the noise production and transmission characteristics of small hermetically sealed compressors would be invaluable for noise reduction. Identification of individual compressor noise problems and effective noise reduction methods could then be developed in terms of the general characteristics.

This paper will combine and generalize the results of several research projects (over as many years) on compressor noise characteristics carried out at the Ray W. Herrick Laboratories, Purdue University. A general model of compressor noise generation and transmission mechanisms is presented for understanding and guiding additional research.

GENERAL COMPRESSOR SOUND SPECTRUM CHARACTERISTICS

The two geometric configurations commonly used in small hermetic refrigerant compressors are the rotary vane and reciprocating piston. Both configurations utilize a progressively reduced volume compression chamber in which the refrigerant is mechanically compressed to a higher density and pressure. A direct drive electric motor is attached to the compressor frame and the compressor system is spring supported in a hermetic shell. From mechan-

ical similarity it could be hypothesized that the two compressor types would have comparable noise generation and transmission characteristics.

Figure 1 shows the sound spectrum of a rotary vane compressor (2) while Figure 2 shows the sound spectrum of a reciprocating piston compressor (3). The two compressors were roughly the same horsepower but had different pumping speeds. The rotary vane compressor had two vanes at 3500 RPM giving a pumping frequency of 117 HZ and the reciprocating piston compressor had one piston at 1750 RPM giving a pumping frequency of 29 HZ.

The sound spectra of both compressors are primarily composed of pure tones (with varying amplitude) of the harmonics of the pumping frequency across the entire spectrum. Detailed examination of the sound spectra will also show some lower frequency harmonics of the electrical frequency and the shaft rotation frequency. The sound spectra of both compressors have bands of pure tone activity and frequently a higher level single pure tone around 500 HZ (Figure 1-460 HZ, Figure 2-570 HZ).

Measurements of compressor noise were made in an anechoic room with a lower cut off frequency of 100 HZ with the microphone at the midplane of and approximately 3 ft. from the compressor shell surface. Comparison with the ambient noise spectrum showed that all tones above 10 dB were attributable to the compressor. Repeatability tests on the rotary vane compressor, to determine control of the experiment, produced 90% repeatability to within 4 dB of all harmonics between any two runs. Better agreement is obtained if levels are averaged over several runs.

COMPRESSOR NOISE GENERATION AND TRANSMISSION MECHANISMS

Figure 3 is a proposed conceptual diagram

of the general noise generation and transmission mechanisms of small hermetic refrigerant compressors (4). The figure depicts the exciting mechanisms acting on the internal members of the compressor and the paths that the resulting noise and vibration must take through the compressor to the hermetic shell and on to the appliance cabinet.

The exciting forces acting on the moving and stationary parts, i.e., piston or rotor and the frame or stator, consist of the inertia forces of the moving parts, the magnetic field forces from the motor windings and the gas pressure forces from the compression chamber pressures. The inertia forces will occur at the lower harmonics of the rotational speed and the magnetic field forces at the lower harmonics of the electrical field frequency. In comparison the compression chamber gas pressure forces will generally be rich in harmonics of the pumping frequency due to the rapid pressure rise in compression and sharp cutoff after discharge.

At equilibrium the compression chamber gas pressure forces will be balanced by the bearing oil film pressure forces and the electric drive motor torque. However, due to time lags in the oil film and the electric motor, this set of forces and torques will not be in balance instantaneously (only in average over one rotation) and will cause vibratory motion of the compressor at the harmonics of the pumping frequency. The vibrations of the compressor will then be transmitted to the hermetic shell through the suspension system and the suction-vibration will disturb the surrounding gas producing gas-borne excitation to the shell.

The compression process and the manner in which the suction and discharge valves operate will produce pressure pulsations in the suction and discharge gas. High side compressors may communicate the discharge gas pulsations to the gas in the shell while low side compressors may communicate the suction gas pulsations to the gas in the shell and produce gas-borne excitation to the shell. Mufflers are standard equipment of compressors to reduce this excitation.

When any of these excitations occur at such a frequency as to excite a resonant condition of the shell or a gas resonance within the shell, the resulting magnification will produce increased compressor noise effects.

COMPRESSOR VIBRATION EFFECTS

Acceleration measurements were made on the rotary vane compressor to investigate

the relationship between compressor vibration and compressor noise (2,4). The resulting compressor acceleration spectrum is shown in Figure 5 (a). Harmonics of the pumping frequency are clearly identifiable up to the 43rd harmonic. While the 460 HZ pure tone is not present (actually a gas resonance), the band of pure tones from 1800 HZ to 2400 HZ found in the noise spectrum is readily apparent. With the exception of the 460 HZ pure tone, comparison between the noise spectrum of Figure 1 and the acceleration spectrum of Figure 5 (a) can generally be made across the total spectrum.

Figure 4 (a) shows typical time traces of the compression chamber pressure at three locations for one half shaft rotation. The chamber pressure increases until the discharge valve opens and the gas flows out. The overpressure pulse occurs due to restrictions in gas flow through the valve system. When the rotating vane passes the discharge valve a sudden burst of compressed gas from the discharge valve port expands back into the following compression chamber causing a pressure ripple on the compression trace.

Figure 4 (b) shows the pressure traces after a modification of the valve system to "soften" the traces and reduce the higher harmonic content of the compression chamber pressure. The modification consisted of increasing the clearance volume and the flow area. The effect was to smooth the overall trace, reduce the overpressure pulse, and eliminate the flash back pressure ripple.

Figures 5 (a and b) and Figures 6 (a and b) show the effect of the modification in producing a reduction of the compressor acceleration spectrum and the compressor noise spectrum. The noise reduction effect was most pronounced above 2000 HZ.

Recent work (5) using coherence analyses between the compression chamber pressure trace and the compressor noise has shown that the pressure ripple has little effect on noise while the overpressure pulse and the general shape of the pressure trace (producing higher harmonics) have a major effect on noise levels.

GAS RESONANCE EFFECTS

The volume of gas contained between the compressor and the hermetic shell will vibrate with resonant amplification if the excitation frequency coincides with one of the natural frequencies of the gas volume. This phenomenon will exist independently of the compressor vibration.

Analysis of the rotary vane shell cavity

assuming the compressor occupied an idealized cylindrical internal volume predicted the first gas resonance frequency should occur around 500 HZ at the normal operating gas temperature (4,6). Experimental study of the 460 HZ pure tone in Figure 1 revealed a strong sensitivity of its amplitude to gas temperature. As the temperature of the gas changes, the sonic velocity and therefore the resonant frequency will change. This temperature sensitivity suggested that the 460 HZ tone was a gas cavity resonance amplification of the 4th harmonic of the 115 HZ pumping frequency.

A test system utilizing two identical rotary vane compressors was used to study the gas cavity resonance effect. Discharge gas from a running compressor (containing all the pumping harmonics) was fed into the hermetic shell gas cavity of a static (non running) compressor. This system provided gas pulsation excitation to the static compressor gas cavity while eliminating the other noise sources of a running compressor. The temperature of the discharge gas flowing into the static compressor was raised so that the sonic velocity and the cavity resonance frequency increased.

The effect of this change in gas temperature and resonance frequency is shown in Figure 7. At the beginning of the sequence (177.5 F) the gas resonance frequency is approximately 10 HZ below the 4th harmonic excitation frequency and as the temperature increases the gas resonance frequency shifts through the constant frequency 4th harmonic. A simple change in the geometry of the compressor was also tested and effectively moved the gas cavity resonance frequency away from the 4th harmonic with the resultant major reduction of the 460 HZ pure tone.

HERMETIC SHELL VIBRATION EFFECTS

Tests on several different compressors have shown the hermetic shell housing to be the principal radiator of noise (3,7,8). The excitation of the shell takes place in two ways; the gas cavity refrigerant gas pressure acting over the inner surface of the shell, and the vibration of the suspension springs and the suction-discharge tubes acting on the shell at the connection points. The resonant characteristics of the shell and the resulting magnification of the excitation at the resonant frequencies can produce major effects in the overall radiated compressor noise.

The resonant frequencies and mode shapes of the rotary vane compressor were determined experimentally (8). The shell was excited with an electrodynamic shaker

through a force transducer. Miniature accelerometers, for low mass loading of the shell, were used to measure the shell motion. Resonant frequencies and mode shapes were obtained for the compressor shell with the compressor mechanism mounted inside.

Figure 8 presents the comparison of the measured shell resonant frequencies to the sound spectrum of the rotary vane compressor. Practically all of the shell resonant frequencies are identifiable in the acoustic spectrum. With the exception of the 460 HZ gas resonance, the major peaks in the acoustic spectrum generally occur at the shell resonances.

CONCLUDING REMARKS

A general concept of the noise generation and transmission mechanisms of small hermetic compressors has been presented with supporting experimental evidence. The following statements review and delineate the presented mechanisms.

1. The major source of the compressor noise is the compression process and its generation of multiple pumping harmonics.
2. The dynamic action of the valves will affect the compression process and the noise generated.
3. The noise is transmitted to the suspension system and the surrounding gas in the shell by compressor motion or by suction-discharge tube openings.
4. Compressor mufflers (suction or discharge) will reduce the noise transmitted to the surrounding gas in the shell by the suction-discharge tube openings.
5. Gas resonances of the gas in the shell will magnify the exciting pumping harmonic with resulting high noise transmission to the shell.
6. Transmission of noise to the shell occurs through the shell gas and the compressor suspension system.
7. The shell is the major radiation of the noise. Resonances of the shell also magnify the exciting pumping harmonics with resulting high transmission and radiation of the noise.

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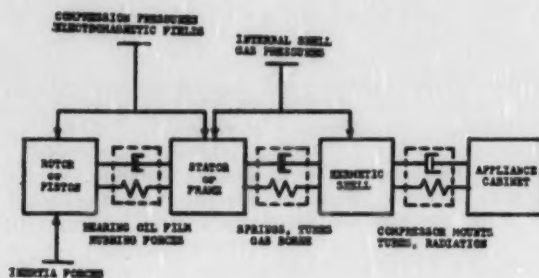


FIGURE 2 COMPRESSOR NOISE GENERATION AND TRANSMISSION MECHANISMS

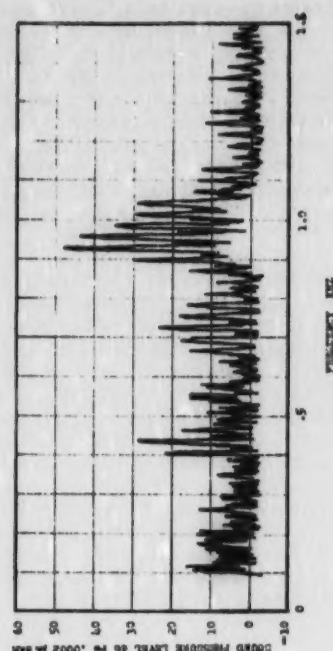


FIGURE 3 NOISE MAGNITUDE SPECTRUM FOR PISTON COMPRESSOR NOISE SPECTRUM

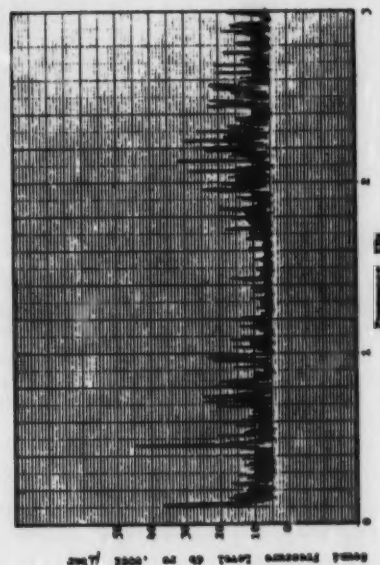


FIGURE 4 NOISE MAGNITUDE SPECTRUM FOR VANE COMPRESSOR NOISE SPECTRUM

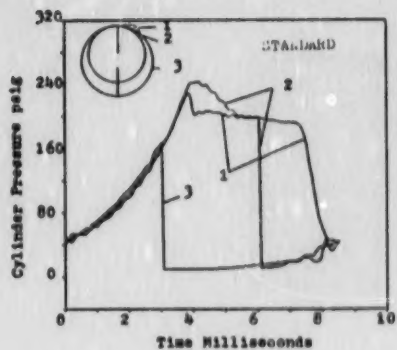


FIGURE 4. ROTARY VANE COMPRESSOR COMPRESSION CHAMBER PRESSURE TRACES

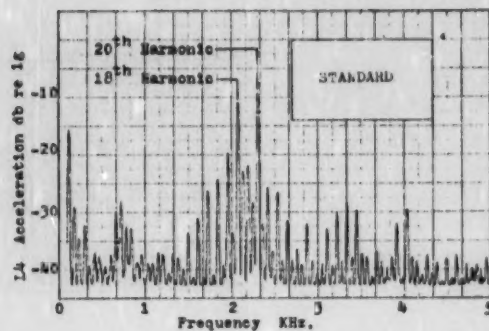


FIGURE 5. 20 HZ BANDWIDTH ROTARY VANE COMPRESSOR ACCELERATION SPECTRUM

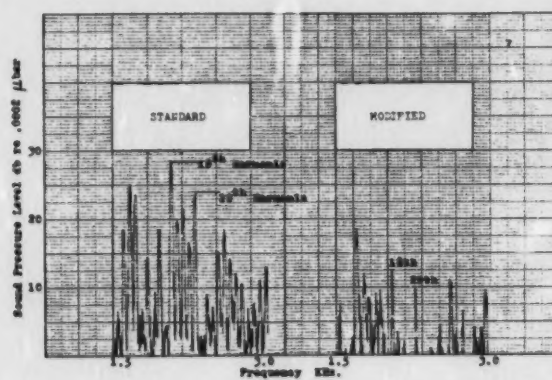


FIGURE 6. 20 HZ BANDWIDTH ROTARY VANE COMPRESSOR NOISE SPECTRUM (PORTION)

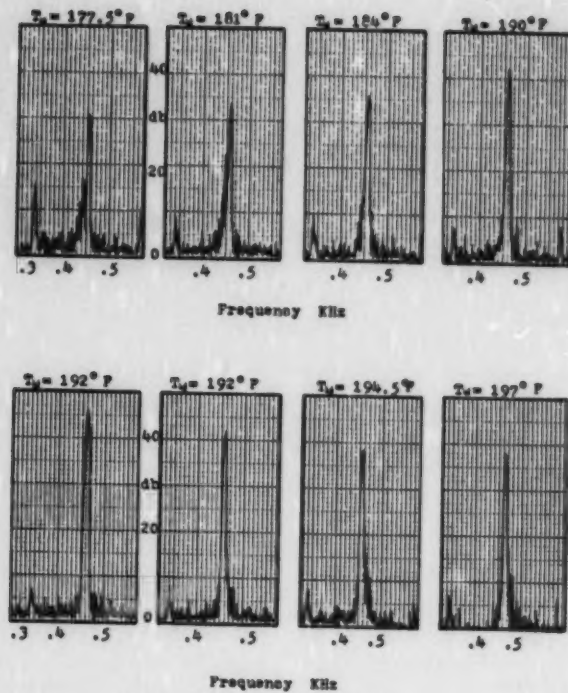


FIGURE 7 GAS RESONANCE IN SHELL CAVITY

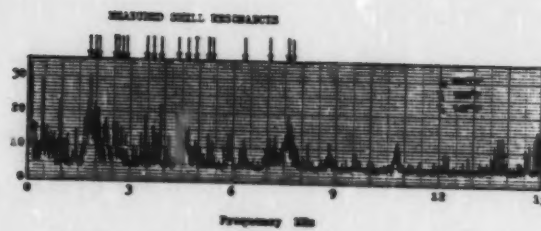


FIGURE 8 ROTARY VALVE COMPRESSOR NOISE SPECTRUM AND SHELL FREQUENCIES

"Noise and Vibration Study of Small
Hermetic Refrigerant Compressors"
presented by
J. F. Hamilton

DISCUSSION:

J.R. Bailey:

In the coupling with shell resonances,
do you feel damping would be effective?

J. Hamilton:

We have put damping on these shells.
The problem is getting a material that will
stand the temperatures and be effective.
Internal damping coatings are completely
out because inside the shell these are
little chemical plants and the manufacturer
would like to guarantee them for at least

5-10 years. On the outside they're a little
fussy about appearance. We have tried spray
on coatings and have gotten a bit over the
4 dB reduction we felt we had to get. Other
manufacturers have tried coulomb damping
treatments inside the shell. These are
effective for a short length of time until
wear sets in, then they degrade. We have
also tried to detune the shell, but this
becomes almost impossible because of the
number of harmonics generated.

FC CENTRIFUGAL FAN NOISE

David Raj, Graduate Instructor
William B. Swim, Professor
Department of Mechanical Engineering
Tennessee Technological University
Cookeville, Tennessee 38501

INTRODUCTION

Forward curved centrifugal fans are used extensively in air conditioning and ventilation systems. These fans produce a significant amount of noise and are frequently the major source of system noise. Due to the small system size and the close coupling of small FC centrifugal fans to occupied spaces, high annoyance to people generally occurs. To avoid size and cost problems resulting from the use of sound attenuation devices, the sources of the noise need to be reduced in strength. A research study has been initiated at Tennessee Technological University to meet this need. This paper describes the work in progress on FC fan noise.

The three most significant contributors to FC centrifugal fan noise are thought to be:

- the rotor noise sources
- the effects of the housing and
- the rotor-cutoff interaction.

The rotor produces broad band noise and is believed to be the largest contributor to the fan noise. The housing appears to act as a resonator and filter which modifies the rotor noise. The rotor-cutoff interaction can produce strong discrete frequency noise. Thus a significant reduction in overall fan noise appears possible only if the strength of the rotor noise is reduced. And rotor noise reduction can best be achieved by identifying and describing the sources of the rotor noise. Then effort can be effectively applied to eliminate and/or reduce the strength of the major rotor noise sources.

Most of the rotor sources appear aerodynamic in origin. So a likely result of the noise reduction work will be a decrease in the intensity or extent of the unsteady flows in the fan. A by-product of the improved flow patterns in the quieter fan design could be a more efficient fan.

BACKGROUND

Studies of centrifugal fan noise have, to this day, been very limited in both number and scope. Most of the research is experimental. Analytical

analyses are nearly nonexistent. Early workers [1] measured fan noise levels and developed empirical fan noise equations. The equations related the noise generated to the fan power input and other fan performance parameters. At best, these empirical correlations provide a fair estimate of the total noise produced by a fan. But they give no clue to the sources nor mechanisms of noise generation. They are hence of little or no use in the study of the noise sources in a fan.

The first attempt to study fan rotor noise was made by Howes and Roal [2]. Their rotor noise measurements were made using a rotor operating without a housing. As an uncased FC rotor will normally operate in stall, their results seem to be for stalled flow. And analysis of noise from a stalled rotor is of limited use only. Noise measurements made by Chanaud [3] on an uncased rotor also gave the noise from a stalled rotor.

Huebner [4] suggested that rotor noise came mainly from the vortex shedding mechanism, giving rise to broad band random noise. A few attempts to reduce the broad band rotor noise have been made in the past. But they have not met with much success. Embleton [5] put a series of transverse slots in the blades. He hoped to postpone the separation of the boundary layer from the low pressure side of the blade. This modification produced some performance loss but little noise reduction. Petrov et al. [6] reduced the broad band noise by placing screens at the entry to and exit from the blades. A considerable decrease in fan efficiency accompanied a small noise reduction.

The pure tone noise at the blade-passage frequency and its harmonics is produced by the rotor-cutoff interaction. A siren type or monopole generation mechanism was suggested by Huebner [4]. But measurements show the source to be dipole in nature. So the blade passing noise likely results from the fluctuating forces on the cutoff. The fluctuations would be caused by the interaction of the cutoff with the strong velocity gradients that seems to occur as the flow leaves the blades. Methods to reduce this source have been studied in detail by many researchers [5,7]. The suggested modifications, to achieve a 10 to 12 dB noise reduction in the pure tone levels, are:

1) increasing the rotor-cutoff distance, 2) sloping of the rotor blades or 3) sloping of the cutoff with respect to the blades. All three modifications reduce the magnitude of the force fluctuations.

The fan housing does not appear to be a major noise source itself. But rather, it acts to amplify or suppress the noise radiated by the rotor. Measurements [8] have shown the enhancement of certain frequencies by the housing. The lowest frequency corresponds to the Helmholtz resonant frequency. The housing essentially behaves like a small reverberant "room" placed around a large noise source. Agnon et al. [9] "softened" the housing environment by perforating the housing and covering it with sound absorbing material. This absorption produced a measurable reduction in the high frequency broad band noise. But the overall noise levels did not drop much.

Hence, apart from the reduction in the rotor-cutoff interaction noise, very little fan noise reduction has been achieved to date. Analytical analyses on FC centrifugal fans have not been published because of the mathematical complexity. Lighthill's noise theory [10], and Curle's modification [11] to include solid bodies, have been applied to simple flow problems with some success. Sharland's [12] analyses of axial fan noise sources showed promise. But some of his results have been challenged for axial fans and his methods have not yet been applied to centrifugal fans. Finally, Yeow's [13] attempts to model the acoustic behavior of a ducted centrifugal rotor were successful at very low frequencies (<150 Hz).

The problem of FC centrifugal fan noise is indeed very complex. Analytical analyses require the input of detailed knowledge of the flow field including second order effects, throughout the fan. This type of flow detail for the complex fan system is impossible at this time. Thus, the problem can best be tackled experimentally.

OBJECTIVES

The rotor appears to be the prime source of noise in a centrifugal fan. The reduction in the rotor noise is the first step towards development of quieter fans. Modifying the rotor geometry to obtain a significant noise reduction can best be achieved after the noise sources on the rotor are identified and the mechanisms of noise generation properly understood. Hence, the objectives of the research are:

- 1) To measure the noise spectra of a centrifugal fan rotor and to identify the important noise sources.
- 2) To apply the information on sources to redesign the rotor to achieve a significant reduction, 10 dB or more, in fan noise levels.
- 3) To determine the effect of the fan housing on the noise generated by the rotor only.

Most of the work is expected to be experimental. Analytical models, however, will be used for an order of magnitude comparison of the noise sources.

THE APPROACH

The fan selected for this project is a single width, single inlet 10" FC centrifugal fan made by LAU Industries. The nondimensional performance curve for this fan, based on manufacturer's data, is shown in Figure 1. Performance measurements made in this study closely follow this curve.

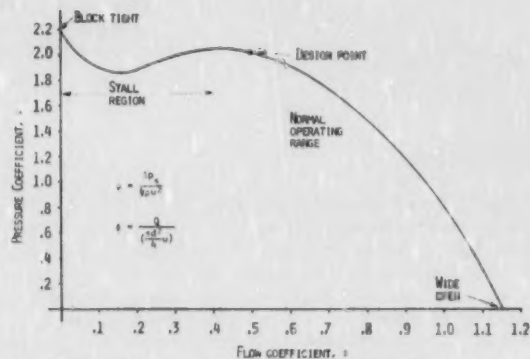


FIGURE 1 NONDIMENSIONAL PERFORMANCE CURVE FOR LAU 10" SMC FC FAN

Experimental Approach

Preliminary experiments are being performed to study the flow in the fan using tufts, smoke and hot wire probes. The test setup for the experiments is shown in Figure 2. The fan is being run

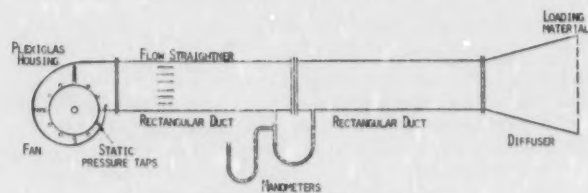


FIGURE 2 TEST SET-UP FOR FLOW VISUALIZATION EXPERIMENTS

at low speed (800 rpm) to make the flow visualization more effective. The housing of the fan was made completely of plexiglass to the same shape and size as the metal housing. Pressure taps are mounted on the housing to measure the pressure distribution around the wheel. To see the tufts and the smoke in the housing, the rotor is "stopped" using a strobe, which is actuated by the rotation of the fan.

Some of the preliminary experiments have been completed and the results confirm some assumptions about the nature of the flow. The pressure distribution around the wheel at various flow rates are shown in Figure 3. The polar plot shows that the pressure is far from uniform. Due to the large

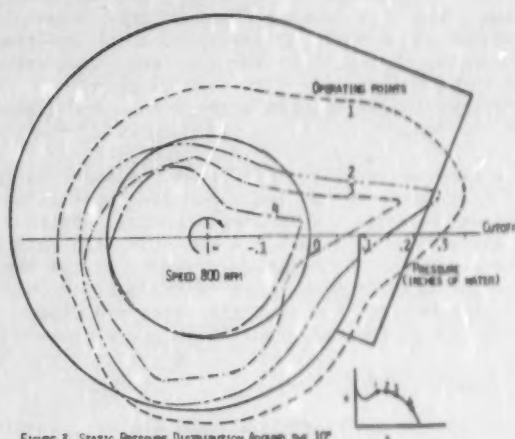


FIGURE 3 STATIC PRESSURE DISTRIBUTION AROUND THE 30P PC WHEEL WHEN OPERATING IN PLEXIGLASS HOUSING

tangential pressure gradients, the flow in the impeller will be a strong function of tangential position. Each section of the impeller will tend to operate at a different back pressure and, therefore, at different "operating points."

The flow entering the blades was found to be far from ideal. At the inlet end the flow was axial and at the back plate the flow was radial. The flow through the blade passage in the region close to the shroud was highly unsteady with back flow. At the blade exit, there was very little through flow in the regions close to the shroud. Also regions of flow separation were observed on the suction side of the blades. More work with flow visualization is being done to get a better description of the flow field.

The velocity of the flow at a few locations will be measured using a hot wire probe. The visual results will be used to select the critical spots for the velocity measurements. The combined hot wire and flow visualization data should give a good description of the flow field.

After the preliminary experiments, two major experiments will be performed.

1. Experiments on rotor noise alone. Efforts in the past to measure the rotor spectra have given the spectra due to stalled flow over the blades. To avoid the problem of rotor stall, a radial diffuser will be put on the wheel. The proposed experimental setup is shown in Figure 4. The pressure recovery of the diffuser should prevent the wheel flow from stalling. The stability level of the flow through the rotor is not expected to change appreciably with the replacement of the housing by the radial diffuser.

The flow at the blade outlet will be measured using a hot wire probe. Static pressure taps are to be

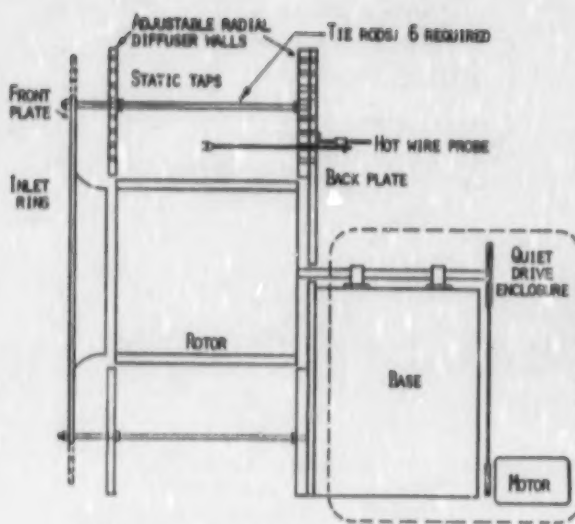


FIGURE 4 RADIAL DIFFUSER APPARATUS FOR ROTOR NOISE MEASUREMENT

located radially outwards on the diffuser plates. These velocity and pressure data will be used to define the flow corresponding to the measured noise levels. The flow patterns will also be correlated with the noise spectra. The noise spectra of the unit will be measured in a reverberation room to expedite the noise measurements.

This setup will be used to measure the rotor noise as a function of through flow rates. The rotor will be loaded by changing the radius--and pressure recovery--of the diffuser and by changing speeds. A study of the changes in noise spectra will indicate some of the noise sources. The same setup will be used to obtain the distribution of the important noise sources in the fan. For this purpose, cross correlation techniques, used successfully in the past for identification of noise and vibration sources, will be used.

The two microphone arrangement to be used for the cross correlation experiments are shown in Figure 5. These measurements will be made in an anechoic room. The microphones will be positioned such that only a small arc on the wheel can be "seen" simultaneously by both microphones. The difference in the time taken by the sound signals--leaving a source on the arc--to reach the two microphones is a function of the axial location of the sound source on the blades. Thus a time delay cross correlation of the signals received by the two microphones will give the relative importance of the sources distributed axially on the blades within the arc "seen" by the microphones. Since the flow is axisymmetric, this distribution should be the same all around the wheel.

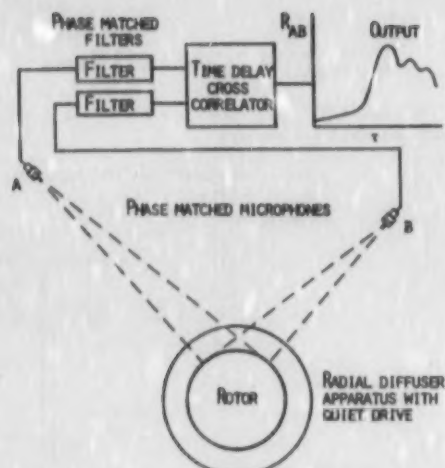


FIGURE 5 SKETCH OF THE MEASUREMENT SYSTEM FOR THE CROSS-CORRELATION EXPERIMENTS (APPARATUS TO BE LOCATED IN ANECHOIC CHAMBER)

2. Experiments with the complete fan--rotor and housing. The noise produced by the fan will be measured using the simple in-duct method. No special test rooms are required. The in-duct method has been found to give reliable fan noise measurements [14]. Details of the procedure are outlined in the proposed ASHRAE Standard 68P [15]. The experimental setup is shown in Figure 6. It

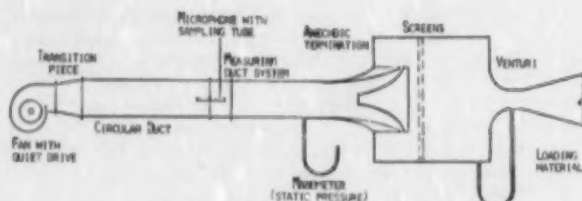


FIGURE 6 TEST SET-UP FOR IN-DUCT NOISE MEASUREMENTS

consists of a fan connected to a circular duct via a transition piece. An anechoic termination at the end of the duct prevents the reflection of sound back towards the fan. The flow is measured downstream of the anechoic termination. Venturi flow meters--with two or three different throat sizes--will be used to minimize the pressure loss in the system.

The pressure loading of the fan will be achieved by placing porous materials on the diffuser outlet. The fan static pressure will be measured with a plezometer ring at a suitable location in the duct.

The noise produced by the fan will be measured using a microphone fitted with a special sampling tube [16]. The tube-microphone combination will be located at an optimum radial and axial position in the duct. Corrections for the frequency response of the sampling tube will need to be made on the measured levels to obtain the actual noise spectrum of the fan.

The flow and static pressure at the blade outlet will be measured--during the noise runs--using hot wire probes and static pressure taps. These measurements will indicate the operating points at the various circumferential regions in the fan. An attempt will be made to correlate the flow patterns in the complete fan with the noise and flows obtained in the radial diffuser experiments.

Flow Noise Models

Even though analytical analyses are very complex, simple flow noise models can be effectively used to guide and interpret the experimental results. An order of magnitude analysis of the various sources can eliminate the weakest sources from consideration. Flow noise models will be used to guide the study of the following sources:

- 1) the jet-wake flow leaving the blades
- 2) separated flow in the blade passage
- 3) turbulent boundary layer noise
- 4) incident turbulent flow over the blades
- 5) siren effect at the cutoff

INTERPRETATION OF RESULTS

The information obtained from the experiments on the rotor source location and strengths will be used to guide the redesign of the rotor. Rotor modifications will then be made to eliminate or reduce the major noise sources. Measurements will then be made on the redesigned rotor to determine the change in the noise level obtained. A noise reduction of at least 10 dB is expected for the "quiet" rotor. It is likely, however, that the redesigned rotor may require a new scroll to get the full benefits of the rotor modifications. The emphasis in this study is placed on the rotor only. A thorough understanding of the rotor noise sources will be a successful realization of the objectives of this research. Follow-up work on housing is planned for next year. Hence, with a "quiet" rotor and its matching scroll, significant reduction in FC centrifugal fan noise can be expected in the near future.

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"FC Centrifugal Fan Noise"
presented by
W. B. Swim

DISCUSSION:

W. Swim:

This work has been University financed but has had a small amount of industrial input. There is one PhD student working on this and we expect around one and one-quarter man years effort by the end of the first phase.

K. McConnell:

In your presentation, I did not see a description of inlet conditions. Have you given consideration to the circumferential area of the fan as opposed to the inlet diameter?

W. Swim:

We are using a standard fan configuration in which the wheel width is 60% of the wheel diameter.

K. McConnell:

So actually you're asking the air to decelerate and then turn 90 degrees?

W. Swim:

Yes. This is the current design for an FC centrifugal fan. It has a very tortuous path. From the inlet the air decelerates into the blades, the air is then whacked by the blades where it then accelerates across the blades. Then all of a sudden the air sees the housing and a large deceleration again.

P. Baade:

I would certainly think that anything that can reduce the noise of forward blade centrifugal fans by 10-12 dB will put yourself into the position comparable to the people who developed the better mouse trap. There is a huge demand for such fans. Do you have any idea how likely it will be to get that much?

W. Swim:

The chances of success are still highly volatile in my mind at this time. I think there is a good chance to get a rotor whose broad band noise is down 10-12 dB. Whether now we can get this into a housing and get the noise down 10-12 dB is problematical, at this time.

K. McConnell:

Another question occurred to me when you talk about noise reduction. Are you talking about inlet noise or discharge noise?

W. Swim:

The total--both inlet and discharge.

Stanley A. Waggoner
Development Engineer
Agricultural Engineering Dept.
University of California, Davis
Davis, California 95616

James F. Shackelford
Assistant Professor
Mechanical Engineering Dept.
University of California, Davis
Davis, California 95616

Thomas H. Burkhardt
Associate Professor
Agricultural Engineering Dept.
Michigan State University
East Lansing, Michigan 48823

INTRODUCTION

Reducing cannery noise is a complex problem and a challenging project. Practical engineering solutions must be found and implemented to lower cannery noise levels at or below the acceptable levels required by OSHA (1). Currently many working areas of the cannery have excessive noise levels (as high as 95 dBA in general work areas and approaching 110 dBA at local work stations). In some instances individual workers are exposed to noise levels from 250 to 300 percent of OSHA maximum exposure limits (2).

Noise levels in the food processing industry are of the same magnitude as noise levels in many other industries, but the canneries have two additional constraints when choosing materials and techniques for noise reduction. The first constraint is high sanitation standards. All materials used in food processing areas must meet FDA requirements as set forth in the Federal Food, Drug, and Cosmetic Act (3). Environmental materials should not deposit any matter on the food or containers, should not be conducive to bacterial and fungal growth, and should resist steam and high pressure cold water cleaning while not absorbing excessive moisture. A material that does not meet these standards may still be utilized by enclosing it in a material that does meet FDA requirements. Materials used must have smooth, cleanable surfaces that will stand up to continual chlorinated cleaning in a hot, humid atmosphere. The food processing environment, through the FDA regulations, excludes the use of many common noise reducing materials such as porous sound absorbing materials, lead products, and wood products. The second additional constraint of greater importance than in most other industries is economics. Most canneries in California must process all of their crops during a three-month harvest period, and this results in four times the amount of equipment compared to a 12-month operation. Thus, a solution that may be economically feasible for a 12-month industry may not be economically feasible for a cannery. Some related industries such as the beverage industry, the frozen foods industry, the drug industry, and other food industries have a similar operation with many of the same sanitation constraints. Many of the noise control solutions found for the canneries can be applied to these related industries. It must be remembered that the economic constraints for these related industries are, in general, not

as great.

The Agricultural Engineering Department of the University of California, Davis, has recognized the high noise levels of canneries along with the sanitation and economic constraints and has set out to aid in a solution by initiating and continuing a cannery noise research study. The third year of the project will be completed in May, 1977. The first year was funded at a low level and kept a graduate student employed 50 percent of the time. The second year was funded by industry to include a full-time development engineer, a graduate student, and three engineering aides. The third year had the same numbers of personnel and was funded again by industry. All three years included faculty supervision, and participation.

OBJECTIVE

The ultimate outcome of this project will be a reduced noise level for the cannery worker. To accomplish this goal, the overall objective of this research project is to develop effective, efficient, and economical noise attenuation techniques to assist canneries in their noise reduction programs. This project is intended to help them comply with both OSHA noise regulations and the FDA sanitation regulations. Research is needed to provide technical knowledge and information for reducing and controlling cannery noise levels which are too high in many plant areas. The project will formulate and promote procedures for minimizing noise-related problems in the cannery environment.

The project includes many areas of noise control:

- 1) Estimation of the magnitude and sources of noise control,
- 2) Researching the state of the art of noise control in canneries,
- 3) Evaluation of available noise-reduction materials,
- 4) Adaption of existing equipment for lower noise levels,
- 5) Development of quieter equipment for canneries, and
- 6) Dissemination of information among canneries.

PROCEDURE

In reducing industrial noise, OSHA regulations give preference to noise elimination through engineering solutions. If that cannot be done, the second method of reducing worker noise exposure should be the use of administrative controls. Third, ear protectors are the last technique to use in reducing noise exposure to the worker.

This project will address the first; engineering solutions for reducing the noise level. Some of the research will involve combining existing technology with the environmental and operational conditions in the cannery to design add-on components for cannery equipment. Other research will involve new technology to make noise control technically and economically feasible. The engineering design will help convert information and ideas into useful hardware. Prototypes will be designed from the research ideas and built using materials that are compatible to the cannery environment. The prototypes will be used to simulate actual cannery operations and will be tested for functional constraints and noise reductions in an anechoic chamber. The prototypes which seem to meet all the canneries requirements and regulations and have significant noise reduction capabilities will then be set up in actual canneries. The pilot programs will test whether the equipment modification will be durable in a hot, humid cannery environment and will meet the canneries expectations for compatibility with their systems and their expectations in noise reduction. Throughout the project significant information will be disseminated among canneries for their own use.

RESULTS AND DISCUSSION

Cannery Noise Sources

The first-year program included noise level measurements which were compiled in several canneries to determine maximum noise levels and to isolate specific noise sources. All noise measurements on actual machines were taken 1 m from the noise source, using a General Radio Model 1933 Sound Level Meter. The range of noise levels and the corresponding sources are presented in Table 1 (2). Table 2 indicates the noise levels at several workers' stations, taken with the microphone positioned at approximately the worker's ear level (2). The listed ranges of dBA noise level readings represent variations of noise measurements corresponding to differences in conditions and differences in canneries and equipment.

State of the Art

After noise sources and their magnitudes were identified, the state of the art of noise control in canneries was researched. Suggestions for reducing and controlling noise in canneries were summarized. Areas of interest included general noise control ideas and specific suggestions for reducing the noise level on specific machines or components. Can handling systems, high pressure air and steam, positive and negative air systems, blanchers, sealers, and vibrating conveyors were areas of literature research. Many of the suggestions have been available for many years, but most have still not been field tested or implemented in the cannery.

With the harsh environmental conditions and strict sanitation and economic constraints, testing of noise reduction materials was essential to determine which materials could be used in the cannery. Studies on noise reduction, sanitation, durability, installation, and cost were of prime interest.

TABLE 1. CANNERY NOISE LEVELS 1 m FROM SOURCE (2)

Noise source	Noise level, dBA
Boiler room	90-93
Large pumps	95-100
Steam vent	100-105
Steam valves	89-93
Centrifugal fan	89-91
Hot breaks	90-92
Cooking area	90-94
Homogenizer	88-98
Can filler	89-95
Can cookers	90-95
Steam can cleaner	91-93
Blower to dry cans	89-93
Cable can carrier	89-95
Gravity can drop	90-98
Full can conveyor	88-91
Waterfall depalletizer	98-110
Snack pack scrambler	96-98
Surging cans to filler	93-95
Glass bottle fillers	89-94
Glass bottle boxers	88-98
Gearbox and motor drive for cooker	93-96

TABLE 2. CANNERY NOISE LEVELS AT WORKER EAR LEVEL (2)

Worker location	dBA noise level
Boiler monitor	89-93
Cooker monitor	92-95
Filler operator	89-95
Sorting belts	89-91
Can line monitor	85-93
Depalletizer operator	90-99

Materials Testing

Noise Reduction-Sound Absorbing Materials. Originally considered to be a major noise control method, sound absorbing materials appear to be modestly effective. Noise level reductions on the order of 5 dBA can be expected for most applications in the processing area. Higher reductions can be obtained with full interior absorption in an enclosure. Although not a total solution, they are useful and may provide the extra noise reduction to help a particular area comply with OSHA standards. Sound absorbing materials are generally quite helpful when used together with barrier materials and should be used inside enclosures for effective noise reduction.

Ten different attenuating materials were tested by the impedance tube method for sound absorption characteristics using the American Society for Testing and Materials (ASTM) Standard Designation 384-58 (4). Due to the porous nature of these materials, hygienic considerations require these materials to be covered. Film facings of various thicknesses were tested continuously-adhered-to, draped-on, and spaced away from the absorption material. A thin film must be used, so that it does not excessively reflect the sound away from the sound absorbing medium. The best sound absorption for a foam-film composite was obtained with a 1-mil

film draped in front of the absorptive surface. When this film was continuously adhered to the surface, the surface reflected more sound and the absorption was greatly reduced. Also, 2-mil films, which seem to act as a barrier, substantially reduced absorption. However, if these continuously-adhered film-foams are used in a machine enclosure, they may provide a better transmission loss through the enclosure walls. In the many materials tested a wide variation in absorption coefficients was observed. Thus, care must be taken to choose an absorption material with high absorption coefficients. A UCLA experimental ceramic foam was also tested and compared well with commercial materials.

Noise Reduction-Vibration Damping Materials. The use of vibration damping materials can reduce noise radiated by vibrating or shock excited metal surfaces and panels. Many commercial viscoelastic materials in paste and sheet form were tested and found to give moderate noise reductions. However, most of these materials are troublesome to apply and have surfaces which are subject to damage and deterioration. One damping method which overcomes these drawbacks and demonstrates good noise reduction consists of bonding a secondary sheet metal panel to the original panel. Silicone RTV or various other types of bonding compounds can be used in 1/8-inch beads two or four inches on center to bond the sheet and to provide optimal damping. The sheet metal sandwich provides superior durability, excellent noise reduction and ease of installation, all at low cost. Fastening foam to the back of sheet metal panels also provides excellent damping, but again is subject to abuse. Commercially damped sheet metal is also available.

Noise Reduction-Enclosure Materials. Where large noise reductions are needed which cannot be achieved by reducing noise levels at the source, enclosures are most effective and economical. The higher the noise reduction needed the higher the cost. However, an enclosure may be cheaper and easier than modifying or replacing the noisy equipment. Depending on the application, types of enclosure materials will vary from a large concrete wall to a thin clear plastic.

A variety of possible barrier materials was investigated. Special emphasis was placed on those with application in the cannery environment. Other materials such as lead sheet were used as comparative standards. The tests attempted to duplicate the practical problems of enclosure fabrication and sound loss through small gaps and openings. This latter problem is important since total enclosure of a moving can line is not possible.

Direct air leaks are detrimental to the effectiveness of an enclosure. For 40 dBA potential enclosure transmission loss, 0.1% open area will reduce the effectiveness by 20%. A 1% open area will reduce the effectiveness of the enclosure by 50%. Thus, leaks should be limited or designed so there are no direct paths out of the enclosure. Also, for small enclosures with no internal sound absorbing material the noise level may rise 20 dBA from sound pressure buildup and the material selected

for the barrier must account for this buildup.

Sanitation. The growth of bacteria and fungus was observed on various noise control materials. Potential cannery materials for absorbing sound, for damping vibration, for enclosures, and for polymer can guide rails were checked for their possible support of microorganisms. The tests were run at conditions optimal for fungus and bacteria growth following ASTM standards. None of the materials tested provided a carbon source for microorganism growth.

Durability-Wear and Abrasion of Guide Rail Liners. An extensive theoretical analysis of wear and abrasion of can rail liner materials has been completed. Also a prototype horizontal can line lined with ten different guide rail materials ran for approximately 2 months to determine actual can rail liner wear. Both theory and actual testing point towards two of the cheapest plastics, high density polyethylene and polypropylene, as possible wear strips for horizontal empty can lines. There may be areas of high impact and high abrasion, where more expensive materials such as acetal should be used, but a majority of the empty can tracks could be lined with these cheaper materials. It appears that no plastics applicable for guide rail use exist which have a thermal expansion coefficient close to that of steel. Thus, differences in thermal expansions must be provided for with proper installation techniques.

Durability-Replacement of the conventional gravity drop with a clear plastic material functioning as an enclosure in addition to being the can guide is possible. Tests were run on acrylic and polycarbonate sheets to see how they would wear when used as this enclosure can guide for gravity drops. Both materials showed excellent wear resistance. Cellulose acetate butyrate may also be a possibility, as well as non-clear polymers.

Durability-Tear and Puncture of Films. Polyethylene, polyester, polyvinyl fluoride, and polyimide films were tested for tear and puncture resistance in thicknesses ranging from 1/4-mil to 2-mil. For performance relative to cost 1-mil polyester was found to be most satisfactory. Polyvinyl fluoride film also has excellent resistance to tear and puncture, but is more expensive. For areas needing greater strength, 1 1/2-mil films could be used, but the sound absorbed may be reduced slightly.

Durability-Steam and High Pressure Cleaning of Film-Foam Composites. These films must be used to cover foams in the canneries. These films must withstand periodic steam cleaning typical of a cannery. Polyester and polyvinyl fluoride were the only two 1-mil films which showed excellent durability both against steam cleaning and 500 psi cold water. Polyimide stood up well against steam cleaning and polyethylene showed good resistance to 500 psi cold water. For performance relative to cost, 1-mil polyester should be used.

Installation-Sound Absorbing Materials. When porous sound absorbing materials are used in a cannery

environment, they must be covered with a thin film for cleanability and sanitation. The film thickness should be less than 2 mil, with 1 mil being preferable, and the film should not be continuously adhered to the foam, but should be draped or partially adhered.

Draping the film over the foam would be ideal. A good approximation of this would be to place the foam baffle in a 1-mil plastic bag. This plastic bag would then be heat sealed at its open end to close off the porous sound absorbing material from the cannery's environment. Many variations of this heat-sealed plastic bag or folded sheet are possible. Then the sealed sound absorption sheets could be placed in a frame which holds the material tight against the sides and the back. This technique allows the film foam to be held tightly in the frame, but lets the film face be limp to allow the sound to pass into the sound absorption material. When large surfaces are needed, the films can be spot-adhered, leaving most of the film non-adhered.

Installation-Vibration Damping Materials. Vibration damping techniques reduce the "ring" or "tinniness" in a sheet metal panel. For the food processing industry the most durable, easiest applied, least-cost solution is to bond a secondary sheet to the primary vibrating panel. This technique gives good noise reductions and can be applied easily in the cannery with readily available materials. With a sheet metal panel the exposed surface wear and abuse would be much less than that of viscoelastic compound exposed surfaces.

In this damping technique a secondary sheet metal panel is strip-bonded with a viscoelastic adhesive to the vibrating panel. The gauge of the secondary sheet metal panel can be the same gauge or a slightly thinner gauge than the primary resonant panel. The bonding compound can be silicone RTV adhesive, or other bonding compounds with good viscoelastic properties are available. It is applied in 1/8 in. beads diagonally across the metal's surface in 2 in. or 4 in. spacings. A perimeter bead 1/4 in. in from the edge should also be applied. This strip bonding provides optimum damping, making full coverage wasteful and costly.

Sophisticated instrumentation is generally not needed to locate a loud vibrating panel. If one's hand can feel the panel vibrating, chances are that the panel is noisy and damping is needed. A secondary light-gauge sheet does not have to cover the entire vibrating surface, but should initially cover 50% of the area of the primary panel, if all edges of a primary panel are fixed. The secondary panel should be placed in the middle of the panel. If most edges of the primary panel are free, then 100% coverage is needed, but this is not common in machinery applications.

Installation-Enclosure Materials. Enclosures can be constructed of barrier materials only or be combined with damping and absorption materials. If sheet steel is used for the enclosure it should be constrained by applying a viscoelastic damping

material. The addition of sound absorption materials inside the enclosure will reduce the noise level by absorbing reverberant sound waves. Plastics in general do not have to be damped, since they are good at not transmitting structural-borne noise. Also clear plastic barrier materials are available when visibility is needed through the barrier. Plastic enclosures may also be more effective by the addition of sound absorbing materials. In general, the heavier the material, the better the sound transmission loss. Also, sound transmission loss increases with increasing thickness. Air leaks are detrimental to the effectiveness of an enclosure and should be minimized for optimum sound reduction.

Prototype Can Handling

General Considerations. Some can handling components reach levels up to 110 dBA. The common noise generator for all can handling components are the cans themselves. In empty can lines the major noise sources are created by can-to-can interaction, can-to-rail interaction, conveyor cable and can-track interaction, and structural vibration. Filled-can lines have the same noise sources as the empty-can lines although the can-to-can noise is reduced greatly. Empty can noise is significantly higher than the noise radiated by the rails, since empty cans have the smaller resistance to vibration. In filled cans the guide rail generated noise will dominate. The essential features of a typical cannery can handling operation were simulated by prototype systems similar to those shown in Figure 1.



Figure 1. Can handling prototype system.

- (a) can elevator and gravity drop,
- (b) gravity drop and horizontal can line.



Noise levels from horizontal can lines range from 85-104 dBA. Thus, in some cases a noise reduction (NR) of 14 dBA may be needed. Many methods of noise control are available for can lines. Some canneries have reduced the empty can noise level significantly by placing the empty can lines on the roof. This excludes the noise from the processing areas; thus, it lowers the noise level for a majority of the workers. Consideration should also be given to reducing the lengths of the can lines, especially when the depalletizer and filler are far apart or when can lines travel above working stations.

Noise from conveyor cable and can-track interaction needs immediate attention. Steel cables should be replaced with nylon-coated cables (NR: 8-14 dBA). Steel sheave bearings should be replaced with plastic sheave bearings (NR: 3 dBA). The replacement of metal sheaves with extruded plastic guides should be considered (NR: 3 dBA). These guides are not needed if nylon-coated cable is used. The noise reductions listed are for isolated machines with low background noise and, when implemented in stages, are not additive. One can achieve, at most, only the minimum noise reduction if the component or machine is not isolated.

Control of can line speed is an excellent method of noise reduction. When can speed is reduced, the can impacting force is reduced which lowers the noise level. If impact speed is decreased by a factor of 2, the radiated noise will decrease 6 dBA. Thus, the cans should be slowed whenever this will not adversely affect the production efficiency of the system.

Eliminating impacts will effectively reduce noise. Two, somewhat idealistic methods are to always have the cans touching or never allowing the cans to touch. In practical implementation of these ideas, an area to allow can impacting may be needed. This area should be isolated away from employee working areas.

Structural isolation keeps vibrations from radiating throughout the entire can line and also is effective in reducing noise. The use of resilient grommets between frame and side rails is beneficial (NR: 3 dBA). Also, using resilient material to isolate vibrating members from the rest of the can line will help reduce noise.

Full enclosures are capable of providing large noise reductions (NR: 5-20 dBA). An air-tight enclosure which dissipates as much energy as possible is needed. Visual inspection and immediate access to can lines are a must. Thus, at least one clear side and an easy-access door must be provided.

The first priority for reducing the noise from horizontal empty can lines is to install nylon-coated cable and plastic sheave bearings. The next consideration would be to implement can line speed or spacing control. If the noise level is still too high, guide rail liners, structural vibration, positioning of the rails, and full enclosures should all be considered.

Horizontal Can Line. Noise levels for horizontal can lines (Figure 2) range from 95-105 dBA; thus, a reduction of 15 dBA may be needed. Horizontal can lines provide a continuous noise source in contrast with the intermittent nature of noise of other can handling components. Thus, a bold step is to eliminate the horizontal can lines and increase the number of direct can lines while slowing the cans to achieve the desirable acoustic effect and the same amount of productivity. Otherwise guide rail linings (NR: 2 dBA) and proper positioning of the rails (NR: 3 dBA with halving of distance) can be helpful. Inserting magnets in the corner pulleys may help. In horizontal can lines stacked one on the other, sound absorbing baffles placed between each layer are helpful (NR: 6 dBA) (5).



Figure 2. Horizontal can line showing cans and guide rails of a 180° turn.

Gravity Tracks. Noise levels from gravity tracks range from 90-110 dBA. Thus, in some cases a reduction of 20 dBA is needed. This need of large reductions may rule out partial enclosure solutions. The noise must either be controlled at its source or a full enclosure with minimum leakage should be implemented.

Many general noise reducing techniques are available for noise control. Decreasing the slope of the gravity tracks will decrease the can velocity, thus reducing noise 6 dBA for each halving of speed. The noise level of gravity tracks can be reduced by controlling can impacting with sensors to keep the track full or enclosures to surround the impacting area. Keeping the impact region as far from employees as possible has obvious results.

The specific noise control solutions that could immediately be implemented to quiet gravity drops are: 1) constructing an enclosure around the existing line, 2) introducing a flap system that would slow, damp, and reduce impact between cans, and 3) a combination can track, enclosure that would replace the existing gravity drop. Each cannery may have different conditions, thus various solutions for quieting gravity drops are needed. These ideas were tested at the University with results shown and now need to be tested in an actual cannery environment.

For placing enclosures around gravity drops numerous materials are available. The enclosure material finally arrived at depends on the requirements. If visibility is needed, a clear plastic should be used on one, two, or all sides. Flexible vinyls are available along with rigid acrylic and polycarbonate sheets. Butyrate, a clear relatively cheap plastic, is being produced in thicker sheets and is now available. If visibility is not needed or wanted on one or more sides, high density polyethylene or polypropylene sheet can be used to provide a cheaper solution. Damped sheet metal is also a good possibility.

Typically a gravity drop would be viewed and accessible from the side or the bottom. Many enclosures are possible including rigid tubes, rectangular shapes with access doors, flexible wrap-around enclosures, zippered enclosures, see-through enclosures, and combination can track enclosures (Figure 3).



Figure 3. The conventional gravity drop is replaced by a clear acrylic track-enclosure.

A flap system consisting of hanging flaps directly in the can path (Figure 4) has been tested. The flaps slow and damp the rolling cans while allowing them to flow smoothly through the drop at or slightly above the filler flow rate. Presently cans come off the horizontal can lines at a controlled speed and then accelerate to the bottom of the track or impact the preceding can. The flap system would slow the cans to the controlled speed,



Figure 4. Rubber flaps mounted on the gravity drop, slow, damp, and prevent impacting of cans.

damp out the ringing of the can, and fall in between spaced cans to reduce can-to-can impact. The can would impact the soft flap instead of the next can. The flaps reduce impacting and non-impact noise levels from 10-18 dBA. This solution might be combined with an enclosure system for a much greater noise reduction.

SUMMARY AND FUTURE WORK

A three-year project is in progress on the reduction and control of noise levels in canneries. The cannery noise sources and their magnitude were determined. The state of the art of cannery noise control contains numerous noise control ideas, but few have been implemented. A wide range of materials were tested for noise reduction, sanitation, durability, installation ease, and cost. The canneries now know what they should or should not use. In the modification of existing equipment in canneries, empty can handling equipment had first priority. Prototypes were constructed for quiet gravity drops, twists, and horizontal can lines and tested in an anechoic chamber. The most practical noise reduction techniques at this time are: replacement of steel cable with nylon-coated cable, a flap system and guide rails in the gravity drops, new noise reducing twist which will reduce the noise level up to 15 dBA, and plastic rings on all corner pulleys.

Future work will include setting up pilot programs in a cannery to test how the prototypes will function in an actual cannery environment. Other research will include air and steam vent noise, glass container handling, and possible fan and pump noise. The University research will end in May, 1977. Hopefully, pilot programs can be set up during the time remaining and could be reported on later.

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"Noise Reduction in Canneries"
presented by
S. A. Waggoner

DISCUSSION:

C. Hurst:

You tested a lot of equipment. Was this donated by the companies?

S. Waggoner:

Well, the equipment you saw in the anechoic room, bits and pieces of this were donated. The elevator we made ourselves.

R. Cohen:

What is the frequency spectrum of the cans?

S. Waggoner:

We tested all sorts of cans: smooth side, beaded side, 70 pound, 95 pound cans, and we found no significant difference in spectrum between these. The spectrum is broad band--extending clear across the board--from 100 Hz to 15,000 Hz.

W. Gatley:

You mentioned that you bonded some thin film to some acoustical material and its performance was not good. How did you do the bonding?

S. Waggoner:

We purchase these foams with films already bonded. The films were continuously bonded. However, we have found that if you just droop the film on the surface of the foam the absorption does not suffer--say it stays rough, the same at 85 percent. But if you take the same film and bond it continuously, the absorption drops to about 45 percent.

THE NATURE OF NOISE SOURCES IN CONTROL VALVES

Gerhard Reethof
Oliver H. McDaniel
Ali Izmit

The Noise Control Laboratory
The Pennsylvania State University
University Park PA 16802

Industrial control valves and regulators are major noise sources in the chemical, petrochemical and other gas- or steam-using industries. The valve manufacturers over the last several years, responding to noise legislation directed at industrial noise, have been successful in developing quieter valve configurations. Most valve manufacturers have also developed a noise prediction scheme uniquely suited to their own particular configuration primarily based on empirical considerations. However, there is a lack of fundamental qualitative understanding of the valve noise generation, transmission and radiation mechanisms, which, in the past, has presented the development of a universal form of a valve noise prediction technique. This paper presents preliminary research results of an effort to examine the fundamental nature of the noise generation mechanisms in control valves. The source of the noise is believed to be the violent, turbulent mixing taking place downstream of the valve restriction, which, in turn, generates both turbulent and acoustic internal pressure fields. The internal wall pressure fluctuations in the pipe provide the forcing input to the pipe wall. This random force field will excite the pipe wall in accordance with the pipe's dynamic response characteristics. The resulting pipe wall vibrations are usually responsible for the radiated sound.

The Noise Control Laboratory at Penn State has been conducting valve noise research under sponsorship of the National Science Foundation and by several valve manufacturers for over two years.

Our basic approach has been to search for insight into the fundamental aspects of valve noise generation and propagation so that valve noise prediction techniques of a universal nature can be developed. Present techniques are highly empirical and specialized for different classes of valves. We are, in addition, searching for methods to design quiet valves.

The research has been divided into the following tasks:

TASK 1 SPECTRA AND CROSS CORRELATION OF WALL PRESSURE FLUCTUATIONS DOWNSTREAM OF A FLOW CONTROL DEVICE

In this phase of the research we performed cross correlation measurements between flush mounted pressure transducers at different axial locations downstream of orifice plates, valves, and other flow restriction devices, coupled to an anechoically terminated 3" pipe. The basic objective of this portion of the study was to determine the forcing function which causes the pipe wall vibration and subsequent radiated noise.

It was determined from the cross correlation that at greater than ten pipe diameters downstream of all devices tested, the wall pressure field is completely dominated by an acoustic pressure field. Near the devices themselves (within 3 pipe diameters) the wall pressure field is dominated by an intense turbulent pressure field, however, this field does not appear to couple efficiently to the pipe wall.

A typical narrow band spectrum of the acoustic pressure fluctuation downstream of a simple orifice are shown in Figure 1*. The pressure ratio was 2.76 and thickness and diameter ratios of the orifice were 0.2. Moderate peaks in the spectrum correspond to cutoff frequencies of various higher order modes in the pipe. The prominence of the peaks varies with the type of device and operating condition. A typical pipe wall acceleration spectrum for the same conditions as in Figure 1 is shown in Figure 2. Most of the vibrational energy (and subsequent sound radiation) is at or near the cutoff frequencies of the internal acoustic modes of the pipe. As a result of this, we believe that design of in-line wall treatment silencers can be significantly improved by optimizing attenuation in the vicinity of higher order mode cutoff frequencies. Techniques are available for optimization of wall impedance for higher order modes in pipes (see, for example, References 1 and 2).

*Figures 1 and 2 are taken from a recent thesis by A.V. Karvelis, "An Experimental Investigation of the Wall Pressure Fluctuations in Piping Containing Simple Control Devices," Ph.D. Thesis, The Pennsylvania State University, August 1975.

TASK 2 CROSS CORRELATION BETWEEN THE VELOCITY FLUCTUATIONS IN THE VICINITY OF A CONTROL DEVICE AND DOWNSTREAM WALL PRESSURE FLUCTUATIONS

This research investigates the effect of the turbulent flow region on the pipe wall pressure fluctuations at several downstream locations. For this purpose, the cross correlation technique has been used between the turbulent velocity fluctuations sensed by hot film anemometry and the wall pressure fluctuations sensed by a flush mounted microphone. Simple orifices were used to simulate actual valves. These orificial-type intense turbulence generators were used because of their simpler theory, which we believe will give a more basic understanding of the phenomena under study. The results obtained show that this technique can be used successfully in the turbulent flow region where compressibility effects can be ignored. In the highly compressible region aft of the orifice in the first 2 pipe diameter locations, the hot-film senses the temperature, pressure, and velocity fluctuations. A much more detailed study of the signal obtained by the hot-film is required before using this technique in those locations. Typical results shown in Figure 3 indicate that the turbulent flow field at 3 pipe diameters downstream of the orifice and at the centerline, is mostly responsible for the pipe wall pressure fluctuations further downstream. This figure is a plot of the maximum value of the correlation coefficient as a function of radial and axial position.

A typical spectrum of the velocity fluctuations two diameters downstream of an orifice plate is shown in Figure 4. Although the downstream pressure fluctuations, which couple strongly to higher order internal pipe modes, are caused by this velocity spectrum, there is no indication that these modes in any way influence the velocity fluctuations in the vicinity of the control device.

TASK 3 THE MEASUREMENT OF SOUND POWER OF VALVES AND PIPING SYSTEMS

In this phase of the valve noise research we are evaluating a method used for estimating the sound power of valves. Several valves and orifices have been operated at various flow conditions and have been vented to ambient atmospheric pressure both directly and with various lengths of pipe attached downstream. The sound power is calculated using the reverberant chamber method. The basic concern with these measurements is whether data obtained by this method can be used to predict the noise performance of a valve in an actual piping system and whether it can be used to study relative performance of various configurations to evaluate quiet valve designs. It was decided that the validity of the method would be questionable if the results varied significantly with different lengths of downstream piping. Results for a 3 inch conventional globe valve are shown in Figure 5. It can be seen from Figure 5 that the results are reasonably consistent regardless of the length of downstream piping.

As a continuation of this phase of the research, we plan to study the effects of various bends such as elbows and tees, in piping systems, with regard to both radiated noise and the internal pressure field.

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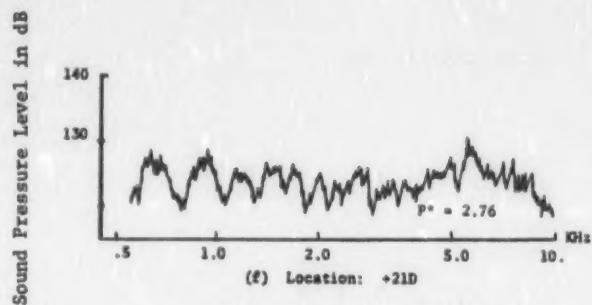


Figure 1

Typical Narrow Band Spectrum of the Wall Pressure Fluctuations for a Simple Orifice

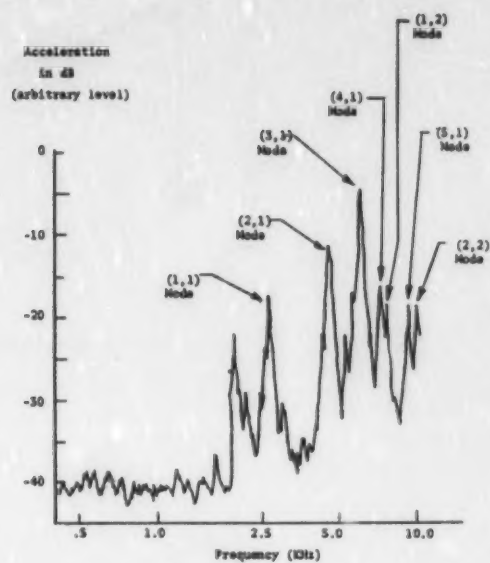


Figure 2

Typical Steel Pipe Narrow Band Acceleration Spectrum for Steel Pipe

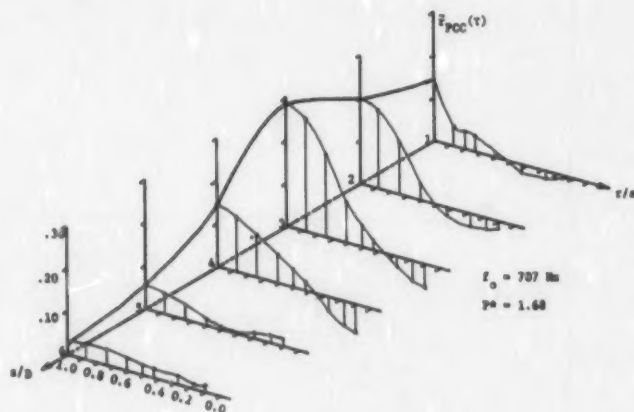


Figure 3

Distribution of the Correlations With Microphone Fixed at $z/D = 16$ and the Hot-Wire Moving Radially and Axially

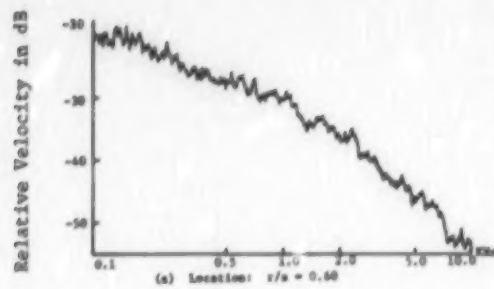


Figure 4

Narrow Band Spectrum of Velocity Fluctuations

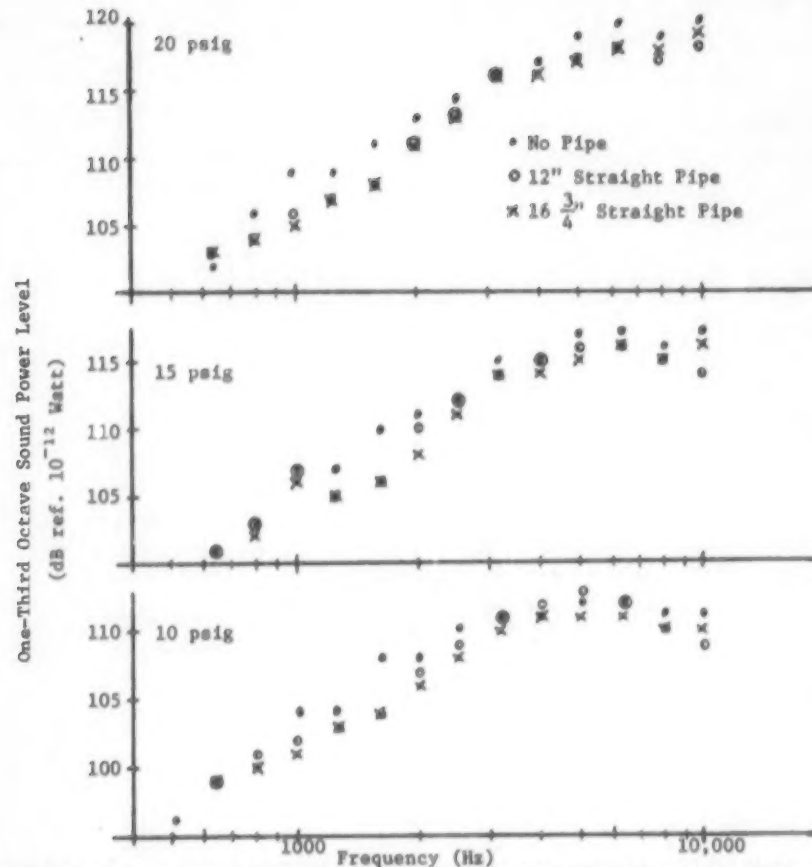


Figure 5 One-Third Octave Sound Power Level Spectra of a 3" Commercial Globe Valve Venting Into a Reverberant Room at Atmospheric Pressure with Various Lengths of Straight Pipe

"The Nature of Noise Sources in Control Valves"
presented by
O. H. McDaniel

DISCUSSION:

O. McDaniel:

This work has been going on for a little over two years. The project started with collaborative funding from several valve manufacturers. This came to about \$6,000. We then submitted a proposal to NSF with strong endorsement of the valve industry through the ISA committee on valve noise. This amounted to two \$50,000 grants for a total of \$100,000. We also have specific contracts with valve manufacturers in product development type work.

P. Baade:

You mentioned that you get peaks in pipe wall vibration at frequencies that are associated with the cut off frequencies of the gas modes, and which do not seem to be identifiable with any pipe wall modes. I think the reason for that is that both the pipe wall modes and the gas modes have cut off frequencies below which they don't exist but which they do exist for frequencies above cut off. For the (2,0) gas mode in which you have two modal diameters, the typical cut off frequency is maybe 2500 Hz for the pipe diameters considered here. If you look at the corresponding mode of a pipe wall as shown in the drawing here (the fundamental hoop mode, elliptical in shape: Ed.), it obviously couples well with this gas mode. If you take a ring of pipe only an inch long, you will find that the natural frequency for this mode occurs at something like one-tenth of the cut off frequency of the gas mode. But now if you don't look at the ring but look at a long piece of pipe you will find that both of these modes propagate at a speed that depends on how far you are above the cut off frequency. They propagate with a phase wave length which depends on their propagation speed. If you look at the equations which describes the propagation of these two modes, you will find that there is typically a frequency just above the cut off frequency of the gas mode where these two modes propagate with the same phase speed, so that they are not only matched circumferentially but also longitudinally. That intersection where they match in all three directions occurs typically at a frequency just above the cut off frequency of the gas mode.

O. McDaniel:

I think we can resolve this by doing careful measurements to find precisely where that peak occurs in frequency.

J. Sneckenberger:

Most valves are fed by a pump of some sort upstream which produces a pulsating flow. Is your study considering these pulsating effects through the valve?

O. McDaniel:

No, we are not studying this. We've taken great pains to see that our flow is not pulsating. We've not considered any interaction effects.

D. Muster:

This is a comment on Pete Baade's comments: In electric motors, there are frequencies excited in the stators of these motors in which you cannot trace to anything, reasonably. Then you find out that there is a harmonic of the stator current which excites this four-pole mode. You have a rotating four-pole magnetic force which excites the four-pole stator mode, so you get somewhat the same mathematical equations, even though the phenomena is quite different.

J. Tichy:

What Peter (Baade) has offered is a very likely explanation, although I would not believe the speed of sound in air would match the speed of sound in the pipe since the latter is much higher. Have you tried to find the modal frequencies of the pipe?

O. McDaniel:

No, but we have plans for doing this. Also we plan to take a look at the correlation between the air and the pipe wall vibrations. Remember though that the phase velocity at cut off is very high--going toward infinity--so there could be a match-up there.

W. Soedel:

The speed of bending waves in a pipe is not the speed of compression waves in the material--it is much, much less.

P. Baade:

If I may be permitted to make a comment on a comment on a comment, the speed of a gas mode just above cut off frequency is much, much faster than the speed of sound. Here you have a wave front which would be parallel to the axis of the pipe and which propagates at the speed of sound. However in the axial direction the trace wave must travel much faster. If you plot

this trace wave velocity against frequency, you will find that it is theoretically infinite at cut off, but then settles down with increasing frequency and approaches the speed of sound. Now if you plot both the speed of propagation of the bending wave in a pipe wall as a function of frequency, and the propagation speed for the corresponding gas mode, you will find a point where they intersect.

L. Faulkner:

We just concluded a study a few months ago looking at transmission through walls of commercial pipes. It was a pipe wrapping study and we did not look at sources. One

interesting thing we found is that if you have a gas inside the pipe, as opposed to a liquid, you get very different characteristics. We concluded that it is the very large difference in impedance mismatch between gas-metal and liquid-metal that explains this. Have you looked at this?

O. McDaniel:

We haven't looked at liquids at all. With regard to valve noise with liquids, there are some drastically different problems.

NOISE CONTROL OF CHAIN SAWS
(A CASE HISTORY)

Werner Soedel, Professor
Ray W. Herrick Laboratories
School of Mechanical Engineering
Purdue University
West Lafayette, Indiana 47907

INTRODUCTION

The work started when a major chain saw manufacturer invited the author to submit a proposal in the area of noise control of chain saws. Before that, three other research projects sponsored by this company had been successfully completed by Herrick Laboratory researchers. The first and the second were concerned with the cutting characteristics of chains, the power that they require, and the general design of two stroke chain saw engines, with the objective of making them lighter. The third was concerned with the reduction of chain saw vibrations as they are felt by the operator. Thus, a good relationship with the company existed, especially one of mutual trust that allowed alterations of the contract objectives as the developing work required it.

The original objectives layed out in the proposal were:

1. To investigate the "noise floor". By this was meant the achievable minimum noise level after elimination of the worst noise sources.
2. To investigate the Power Saw Manufacturer's Association practice of measuring chain saw noise outdoors.
3. To develop an indoor measurement procedure for research and development purposes.
4. To use spectral analysis to identify the major noise sources.
5. To attempt to reduce the general noise level on a saw furnished by the company.
6. To provide continuous advice to company design engineers.

The original time table called for the work to take 14 months. A master's degree candidate was to do the work under the guidance of the author, in partial fulfillment of his degree requirements. The final report on the work was to be a master's thesis. The contract cost was set with this objective in mind.

Fortunately, from the final result viewpoint, a suitable master's degree candidate was not to be found, while a Ph.D. candidate, B.R.C. Mutyala, was available. Unfortunately, from a financial support viewpoint, a fourteen month program does not allow a student to satisfy the much more difficult Ph.D. dissertation requirements, even if he had finished most of the course work, as it was the case with B.R.C. Mutyala. Thus, it was clear from the start that the student was to satisfy the contract first in the allotted time span and then was to pick a particular question for an indepth study for his dissertation.

This is exactly what happened. During the course of the contractual work, the by far dominating noise source was found to be the engine exhaust. The spectral analysis led to the belief that Helmholtz resonator effects involving the engine combustion chamber created most of the noise. Choosing this for the dissertation work, a mathematical proof in form of a noise radiation simulation program of the two stroke engine exhaust noise was the state of the art advancement that fulfilled the degree requirements. Dr. Mutyala received his Ph.D. degree in December 1975. All work, after the initial contract was finally terminated in early 1974, was unsponsored and the student's support came from either Herrick Laboratory fellowships or from teaching assistantships.

The sponsor continued contractual research with the Herrick Laboratories, but in the

area of garden tractor noise control. There was obviously, from their point of view, no profit in sponsoring a continuing theoretical study, whose chance of success, and of becoming practically useful, was small in the beginning. As it turned out, the results of this work became very useful for defining criteria for muffler design.

All this points out a fundamental disadvantage in working with private industry (perhaps excusable in light of all the advantages). It is many times impossible to obtain money to pursue a question in depth, once the immediate problem that caused the company to sponsor research is solved. This is not to say that the relationship with such "utility" sponsors is not excellent and beneficial to a university laboratory, in spite of this difficulty. The relationship with the sponsor has continued to be excellent and of mutual advantage to the present day.

The following describes results of the total three year research as it was recently completed. It is divided into four parts. The first part discusses the difficulties of the industry to get consistent noise measurements and shows that the chain saw or the microphone should be placed on a hard surface in order to get definable data. It is proposed that this procedure should be adopted as a standard.

The second part suggests a procedure for taking indoor measurements using an anechoic room.

The third part discusses noise identification on the chain saw and shows that engine exhaust noise was the dominant feature on the saws that were studied. It shows that Helmholtz resonator type discharges, with the combustion chamber acting as a resonance volume, are a key noise source.

In the fourth part, a design procedure for a simple low pass filter muffler is given that allows to eliminate much of the exhaust noise and brings the saw noise down to acceptable levels.

Finally, a brief discussion of a computer simulation model of two stroke engine noise emission is given.

OUTDOOR MEASUREMENTS

At the time this work was done, no industry standard for measuring chain saw noise was in existence (there may still not be one, for all that the author knows). In an informal way, agreement was reached among companies to measure noise at 50 feet distance, the saw being held waist high and the microphone being located at

ear level. It was quickly realized that measurements were not repeatable, even when it was specified that measurements should be taken over grass turf. The author remembers having seen an informal report that described a sort of round robin experiment where several companies measured each others chain saws, without much success. The results differed by up to 5 dBA.

Rather than trace through all the details of this experiment, which would have required a lot of potentially useless work since the report did not explain the setups in detail, it was decided to approach the question from the viewpoint of fundamental acoustics. The following is a discussion of the conclusions that were reached.

The potential factors that influence outdoor noise measurements and make them appear to be non repeatable are spherical divergence losses, the presence of walls or trees, effects of wind and temperature gradients and the presence of the ground.

We may safely assume that spherical divergence losses are 6 dB for each doubling of distance, provided we take far field measurements, that is we are approximately more than three times the largest chain saw dimension removed from the center of the saw when we take our measurements. Thus, if we take a measurement at 50 feet distance, we will find that the sound pressure level at 100 feet is 6 dB lower.

Now let us suppose we have a lonely wall at 100 feet distance. As a matter of fact, let us suppose the worst possible case: a wall that extends toward infinity in all directions. If we forget at the moment the additional presence of ground, which we will discuss later, we see that the microphone shown in Figure 1 will measure two sounds: The direct sound we want to measure and the sound reflected from our postulated "Chinese Wall". The total sound pressure is a summation of the two effects. Let us take as an example $L_1 = 50$ feet and $L_2 = 100$ feet. Then the magnitude of the reflected sound pressure is approximately one fifth of the magnitude of the direct sound pressure, neglecting interference effects. Thus, a building 100 feet from the microphone will add roughly 1.6 dB to our measurement. The worst case, where the microphone is located directly at the wall, gives us a 6 dB addition. If we postulate, that we can allow a 1 dB error, the required distance between microphone and building is 185 feet.

Note that all reasoning applies also to the case where we interchange the location

of the saw and the microphone.

If the round robin experiment discussed in the introduction had a large enough field in each case is not known.

A more important consideration, often neglected by the practitioner, is the presence of ground. Both saw and microphone are located close to ground, thus similar to the previous reasoning we expect a sound pressure level increase due to the ground being an infinite wall. If either or both microphone and saw are directly at ground level, our measurements show a 6 dB increase as compared to free field measurements (the saw is located in free space or an anechoic room). Again, this is of no consequence since industrial standards can take this automatically into account. What is important, however, is that in certain frequency bands we get interference effects because waves received by the microphone directly travel a shorter distance than waves that reflect from the ground.

For an "acoustically hard" surface like cement or asphalt, we can calculate these interferences. Referring to Figure 2, we obtain "critical frequencies" at which sound is extinguished. If

$$h_r \ll L_1$$

$$h_s \ll L_1$$

then the first of this critical frequencies occurs at

$$f_o = \frac{cL_1}{2h_s h_r}$$

where c = speed of sound.

A typical sound pressure level correction with regard to the free field is plotted in Figure 3. It is also shown that an "acoustically soft" surface will lower this critical frequency. Thus, if measurements are specified to be carried out on grass, the critical frequency will depend on grass length, density, moisture content, etc.

Therefore, if the round robin experiment was carried out over uncontrolled turf, a large source of error existed, especially if the interference frequency was in the main range of chain saw noise. Prodded by this work, there was talk about using artificial turf, but even that will cause problems because of the interference frequency being too low.

It was therefore recommended to carry out experiments on hard surfaces of cement or

asphalt and to place either the saw or the microphone on a pad on the ground (Figure 4). The correction curve is shown in Figure 5. This seems to be the only feasible way to obtain repeatable results that are meaningful to engineers. Nonprofessional certification standards by government and municipal authorities, if they exist, should be corrected to relate to this recommendation.

INDOOR MEASUREMENTS

Outdoor measurements as described can be used for comparison measurements, directional radiation measurements, spectrum analysis and component participation identification. Unfortunately, they did not allow easy identification of the role of engine exhaust since the exhaust noise had to be eliminated from measurements in order to assess possible noise level improvements by an "ideal" muffler. This question had to be answered by measurements in an anechoic room or reverberant room. The anechoic room was chosen since it would best relate to outdoor measurements, when directionality is important.

Measurements in an anechoic chamber required some modifications of the chamber. Exhaust gas had to be removed and the absorbing wedges had to be protected against gasoline and oil pollution. A shroud of plastic sheeting was used for this purpose. One end was open to insert the chain saw and the other end was connected to a tube connected to an exhaust fan (Figure 6).

The plastic shroud, even while thin, caused some loss in measured sound power. For limp plastic sheeting, the transmission loss for sound waves propagating perpendicular to the sheeting can be calculated.

For 2 mil Vis-Queen plastic, the correction curve for transmission loss at random incidence is shown in Figure 7.

Measurements were taken with an anechoic floor. If outdoor and indoor measurements are to be compared, we have to add to the indoor measurements 6 dB for ground correction, subtract shroud transmission loss and subtract 6 dB for each doubling of distance, since inside the anechoic chamber the microphone has to be located closer to the saw by necessity. Such corrections work usually very well and good agreement between indoor and outdoor experimental results was obtained.

To eliminate the exhaust noise from the measurements, the exhaust gas was piped out of the anechoic chamber during a separate experiment. A pipe was attached

directly to the exhaust porting of the engine (Figure 8). Care had to be taken that the presence of the pipe, which was attached to an exhaust fan, did not alter the engine characteristics appreciably. Also, the pipe had to be noise insulated, so that exhaust pulsation excited sound radiation of the pipe did not interfere with the experiment.

Since an operator controlled the speed of the saw manually inside the anechoic room, good ventilation and protective equipment had to be provided.

Because of the difficulty of controlling the speed of the saw for any prolonged time, it was advisable to employ either a real time analyzer or a tape recorder. In the latter case, a relatively short noise recording was played back continuously by splicing the tape into a loop.

SOURCE IDENTIFICATION

Once the capability of producing repeatable frequency spectrums of the saw noise was achieved, the identification of the major noise sources of the saw was possible. Potential candidates were exhaust noise, intake noise, chain noise, cutting noise, structural noise, fan noise, and bearing and piston slap noise.

It has to be remembered that in order to cause an audible change in noise of any significance, there is no sense in wasting ones time with the elimination of secondary noise sources and peeks. Rather, the most important noise source has to be eliminated first. For instance, if there is a noise source generating 80 dB at measuring location and a noise source of 70 dB, the total sound pressure level is 80.5 dB. Eliminating the secondary source of 70 dB completely (which may be an enormous engineering achievement) will reduce the total SPL only by 0.5 dB to 80 dB. However, if we are able to eliminate even only 10 dB of the primary source, the total SPL will be 73 dB.

The primary noise source of the two models of chain saws under study was definitely the exhaust noise (Figure 9). This was of course not surprising since the spark guard doubled as muffler, with more attention usually given to the spark arresting ability than to the muffling ability. But even later, when a good muffler had been designed, the exhaust noise still dominated. Chain noise was found to be insignificant in comparison by running the saw with chain and without. The worst noise level was found by running the saw at cutting speeds with no actual cutting. The cutting process loaded the engine and reduced the total noise output somewhat. This particular reduction

mechanism is not yet completely understood.

The noise due to engine exhaust without a muffler had a definite peak at a frequency that was identified as a resonance frequency of the instantaneous mass in the exhaust ports and the elastic behavior of the combustion chamber. Thus, the engine was found to act like a charged Helmholtz resonator. It was found to be, for a single cylinder two-stroke engine,

$$f_R = \frac{c}{2\pi} \sqrt{\frac{S_p}{L_p V_c}}$$

where

- c = speed of sound [m/sec]
- S_p = effective port area [m^2]
- L_p = effective port length [m]
- V_c = combustion chamber volume at bottom dead center [m^3]
- f_R = resonance frequency [Hz]

The effect produced is pretty similar to the noise produced when opening a bottle charged with pressurized gas. The gas escapes in form of a sudden step. This excites the resonant system.

NOISE ATTENUATION BY MUFFLER DESIGN

The identification of the charged Helmholtz resonator effect was made during the first year of the project. Using a simplified approach, it was found that a simple expansion chamber and tailpipe muffler could be designed, again on the Helmholtz principle, that would work well if its resonance frequency, given by

$$f_{MR} = \frac{c}{2\pi} \sqrt{\frac{S_T}{L_T V_M}}$$

where

- c = speed of sound [m/sec]
- S_T = effective tailpipe area [m^2]
- L_T = effective tailpipe length [m]
- V_M = muffler volume [m^3]

could be made such that

$$f_{MR} < f_R$$

By interaction with the sponsor, it was found that if f_{MR} was very much below f_R , the muffler would work very well as a muffler, but an unacceptable drop in available brake horsepower would also result. If f_{MR} was close to f_R , brake

horsepower losses would be acceptable or even absent, but the muffler would not function very well. Rule of thumb relationships were worked out to give a good compromise (The computer simulation model that will be discussed next helped in this respect).

Just in passing, single volume mufflers with tailpipe had been produced by muffler manufacturers in the past and sold to chain saw manufacturers. It was found that in many cases these mufflers were quite unacceptable. There were cases where $f_{MR} > f_R$, which made the muffler clearly ineffective as far as the most objectionable noise band was concerned. In other cases, horsepower losses were too high.

The muffler designed as part of this project is now in production and is sold as standard saw equipment. Its performance at a typical speed is shown in Figure 10. The old spark guard type muffler was only a few dB below the no muffler case. The reason for this was the complete absence of a tailpipe.

MATHEMATICAL DESCRIPTION OF EXHAUST NOISE

As mentioned in the introduction, it was felt that the straight forward noise control project, in spite of its obvious practical success, was not suitable by itself for a Ph.D. dissertation. Thus, the author gave B.R.C. Mutyala the assignment to develop a mathematical model of the system that would explain the exhaust noise emission behavior of a two cycle engine first without and then with a muffler. The author had worked with combinations of Helmholtz resonator and thermodynamic models before, and had reason to believe in the success of such an approach.

The resulting model consisted of kinematic equations, thermodynamic state equations, mass flow equations, and acoustic equations. The work was published and details can be taken from references [1,2]. A sketch of a typical two stroke chain saw engine is shown in Figure 11.

The following results are for an engine speed of 6000 rev/min at part loads and at full load. Computed typical mean pressures in the crankcase, the combustion chamber and the muffler expansion chamber are shown in Figure 12 for a part time load of 1.0 horsepower.

Figure 13 shows typical measured and computed expansion chamber pressures, at zero load condition of an engine-chain saw system and at an engine speed of 6000 rev/min. Zero chain saw load corresponds

to a part load on the engine between 0.7 and 1.0 hp. While the quantitative agreement is adequate if one considers the necessary idealization of the odd geometries of the real life two-cycle engine connecting passages and the typical uneven firing behavior of a small two-cycle engine, the agreement in oscillatory characteristics has to be termed excellent, especially the frequencies of oscillation.

The experimental as well as the theoretical results show that shortly after the exhaust port has opened, the pressure in the muffler reaches a peak. However, at this time the effective mass plug in the exhaust port is in its extended position and is being forced back by the elasticity of the combustion chamber volume. This high frequency oscillation continues, but decays because of energy loss. By the time the exhaust port closes, the high frequency oscillations has disappeared. What is left is the low frequency oscillation controlled by the expansion chamber and tailpipe. This low frequency oscillation is also visible during the time the exhaust port is open.

The expansion chamber pressure oscillations were computed for the total range of load conditions up to full load. It shows that the same burst like oscillatory discharge characteristic controls the result.

Computed peak pressures in the expansion chamber were calculated as function of load conditions. As expected, these peak values rise with horsepower since an increase in indicated work per cycle produces an increase in combusted gas pressures. This means that the "champagne bottle" is charged with higher pressures before it discharges (before the exhaust ports open). A very simple model that assumes a sudden exhaust opening to full flow area values predicts a linear increase with horsepower and much larger oscillation amplitudes. The more realistic model used here shows smaller amplitudes because the input to the Helmholtz system is of the nature of a ramp. This ramp is strongly dependent on mean flow conditions and exhaust slot shape, or, in other words, on how fast the combusted gas is exhausted.

Figure 14 shows a plot of sound pressure level versus frequency, measured and computed six feet from the open engine exhaust (muffler removed) of a hand-held chain saw. The peak values occur at about 1000 Hz. The experimental points represent one-third octave data, with a spark guard attached to the exhaust. This guard did not influence the sound radiation appreciably. Agreement is excellent,

especially when one considers the poor results that the conventional approach furnishes, where the Helmholtz effect of the combustion chamber-exhaust port combination is neglected. Also no attempt was made to remove sound radiated from the engine and chain saw surfaces.

When the low pass filter muffler is attached, the sound pressure level calculated and measured at the same distance of six feet from the tailpipe end is given in Figure 15. Agreement is again quite satisfying, considering that one would now expect a more visible contribution from surface radiation effects. This is apparent at frequencies of 2000 Hz and higher.

These results show that it is now possible to actually predict the noise emission due to the exhaust of a two stroke engine with and without expansion muffler. It is now possible to design a chain saw muffler during the drawing board stage of design with excellent chances of success.

SUMMARY

This paper attempts to summarize both the results as well as the financial and manpower history of a three year noise control project. The work started as a fourteen month contractual research project that was sponsored by a leading chain saw manufacturer. Good practical results were obtained during this time, but the real state of the art advancement came during the later period of the project, when the work was unsponsored.

Technical results were a repeatable outdoor measurement procedure, a repeatable indoor measurement procedure, noise source identification, muffler design and engine exhaust noise emission simulation.

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2. Mutyala, B.R.C., "A Mathematical Model of Helmholtz Resonator Type Gas Oscillation Discharges of Two-Cycle Engines" Ph.D. Thesis, Purdue University, December 1975.

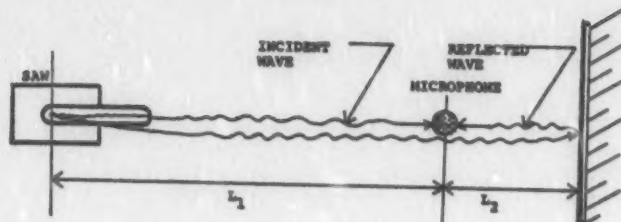


Figure 1. Wall Reflection

Figure 2. Ground Reflection

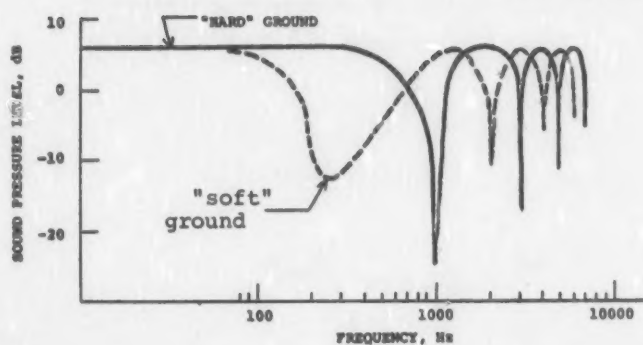
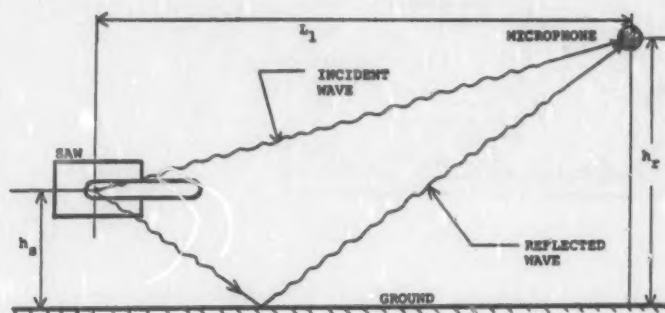


Figure 3. Ground Reflection Correction

Figure 4. Proposed Procedure

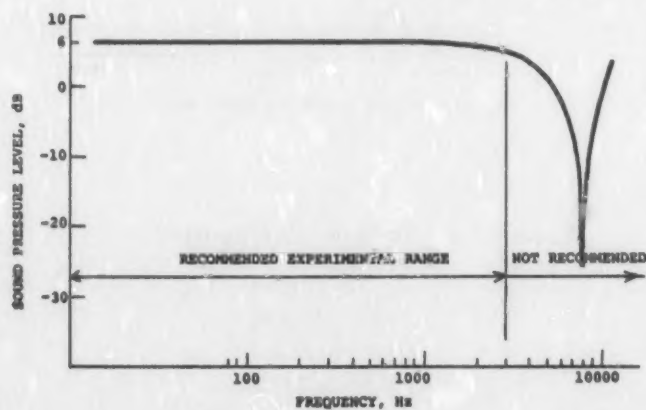
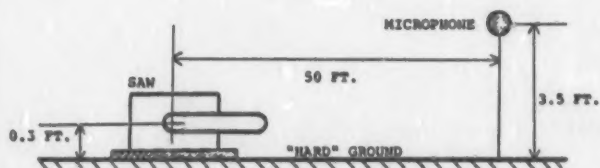


Figure 5. Experimental Range

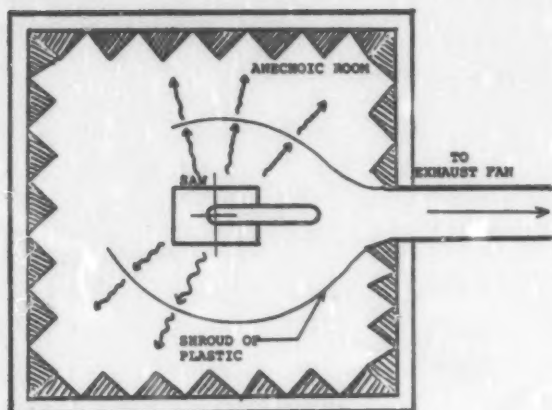


Figure 6. Shroud in Anechoic Room

Figure 7. Shroud Correction

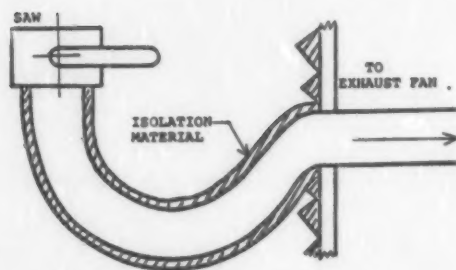


Figure 9. Exhaust Noise Contribution

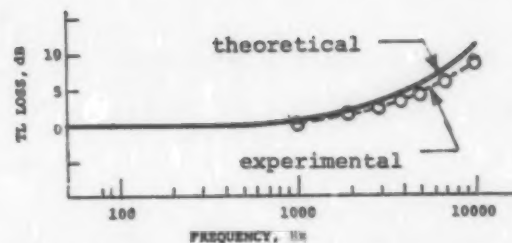
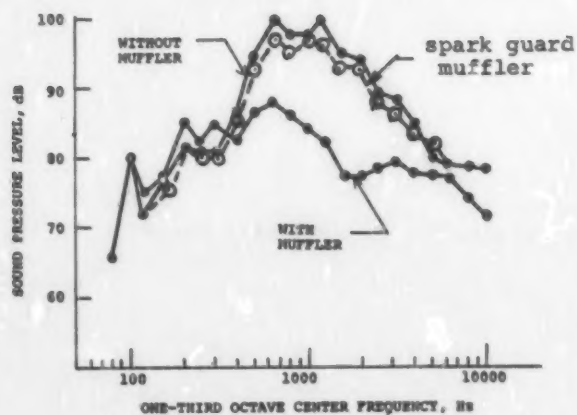


Figure 8. Exhaust Removal

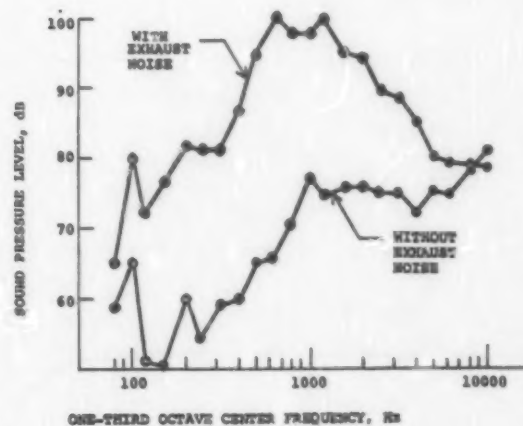


Figure 10. Muffler Performance

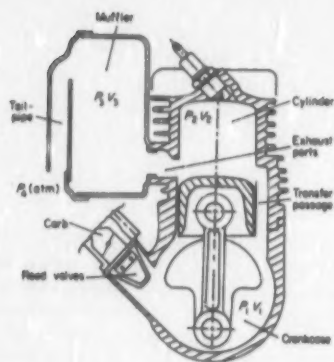


Figure 11. Chain Saw Engine

Figure 12. Thermodynamic Results

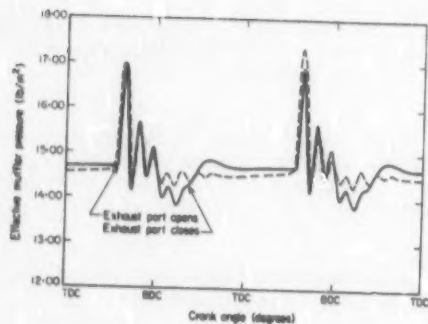


Figure 14. Noise Emission Without Muffler
(● = experimental data)

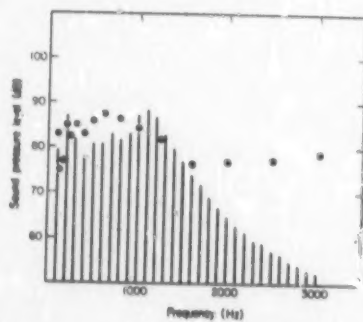


Figure 15. Noise Emission With Muffler
(● = experimental data)

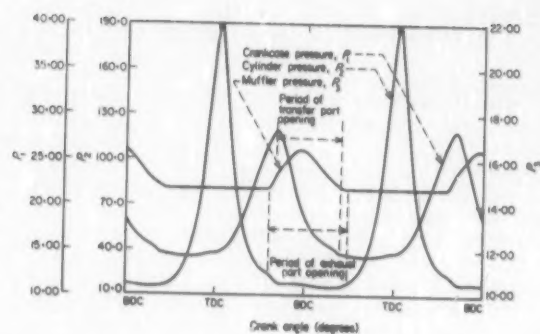
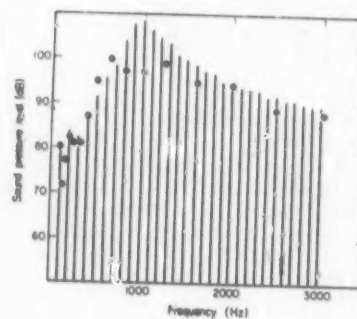


Figure 13. Expansion Chamber Oscillations



"Noise Control of Chain Saws"
presented by
W. Soedel

DISCUSSION:

J. D. Gibson:

In your last figure, what was the fundamental exhaust frequency?

W. Soedel:

The fundamental was around 1000 Hz.

J. D. Gibson:

Does that correspond to the first or second peak shown there?

W. Soedel:

We find only one peak, we don't know where the second comes from. If you look at Figure 14, the experimental data shows two peaks but these two are centered on the theoretical peak at 1000 Hz. We have not refined the model enough to show where this split comes from.

J. D. Gibson:

My second question is: What were the primary sources of noise other than exhaust?

W. Soedel:

It seemed to be radiation from the muffler walls due to pressure pulses. The chain was not important at all.

J. D. Gibson:

I was somewhat surprised by the large band width (in Figure 14).

W. Soedel:

It is not surprising because it is not just one frequency. The ports open and close as the gas discharges. So the port has zero crosssection when closed but when gas is discharged, you have a variable crosssection and variable volume. If you calculate the Helmholtz resonator frequencies, you really get a whole band.

L. Faulkner:

We've done some work on small 4-cycle engines looking at performance effects by scavenging. We found that there is a pressure oscillation effect on performance. If you take, say, the piston at the mid-point and calculate the Helmholtz resonator effect, you can almost predict the pressure pulse in the exhaust ports. Did you find that effect?

W. Soedel:

We also looked at 4-cycle engines and have observed that effect although we have not pursued it because no one has given us the money to do so.

H. Nozick:

Did I understand you to say that the noise level data you indicated was with the load on the blades?

W. Soedel:

All tests were done with an unloaded saw, because we have found that with cutting, the noise level actually goes down a little bit.

W. Gatley:

How big was that muffler you finally proposed to the manufacturer?

W. Soedel:

I don't have the exact dimensions, but it was about this big (showing with hands: Ed.) However, it turned out to be larger than what they were using but much smaller than what they thought the would need.

SOUND POWER MEASUREMENT
AND NOISE SOURCE LOCATION OF LARGE MACHINES IN SITU

Dr. Thomas H. Hodgson

Center for Acoustical Studies
North Carolina State University
Raleigh, North Carolina

ABSTRACT

The establishment of a noise labelling code for large machines situated in the usual reverberant and confined workspaces using the sound power level can be difficult with conventional measurement methods. This is due, in part, to the complicated vibrational motion of the large surface areas encountered in large machines as well as the differing acoustical conditions surrounding various components of the machine.

A sound power measurement method is described which is based on the absolute result for the sound power W radiated by a vibrating surface of area S , namely that $W = \int I_n dS$ where I_n is the surface acoustic intensity and is normal to the surface. Also, the method may be used for noise source identification, where the integral is then taken over each individual component separately. The method avoids all of the problems of the contamination of the pressure field around the machine caused by surface reflections, proximity of machine components, and reverberant effects which normally make the calculation of sound power and noise source identification from sound pressure level measurements a difficult procedure.

An example of the method as applied to a large centrifugal chiller is described using a small accelerometer-microphone transducer combination for the surface acoustic intensity measurements. 'A'-weighted power levels were obtained for components of the machine and the overall 'A'-weighted power level was obtained by simple summation. Although presently no absolute measurements for this type of machine exist, the feasibility of the method was checked by measurement of the acoustic radiation efficiency. In addition, the two dominant noise sources were determined and the variation of power level with frequency was obtained.

The previous and continuing research effort on this sound power measurement method is documented, which includes the development of a rugged transducer arrangement and an evaluation of the minimum number of measurement points required.

INTRODUCTION

Presently there is a need to develop realistic test procedures for the measurement of the acoustical characteristics of large machines as installed in typical reverberant industrial machinery rooms. Where the sound pressure level around the machine can vary greatly depending on the workspace, the sound power level is likely to have a much smaller variation and better describes the large machine as a noise source. Until comparatively recently, it was thought that noise measurements which would characterize the machine and be useful for predicting the noise in other installations could only be made in acoustic test chambers. However, both economic reasons and size prevent the assembly of a large machine in such facilities so that acoustical evaluation of the machine must be accomplished in situ. Fortunately, current thinking has suggested that close-up measurements in situ might be used to obtain meaningful data to characterize the machine and even to determine total noise power. Two sessions at meetings of the Acoustical Society of America in April, 1972 and November, 1974 were

devoted to in situ measurement topics. The application of the absolute sound power measurement technique described here was first presented at the second of the meetings, see Hodgson.¹

The particular machine chosen for this study is a large centrifugal chiller previously described by Ebbing² and is shown in figure 1. Currently it is of interest to know either the overall sound power output of this type of large machine in situ or, for noise control purposes, the sound power contribution of individual components. Then a knowledge of the sound power output of the noisiest components would also give an estimate of the total sound power output by summation and is the approach taken here. At the present time it is known that no method based on sound pressure level measurements alone can estimate the sound power output of this type of machine as installed because of the very complex nature of the machine vibrational motion and the resulting acoustic field. Because the machine surfaces are so large, the total sound radiated is, in general, a complicated function of the surface motion. Thus it does not immediately

follow that the machine component with the largest near-field pressure or the largest motion is the primary noise source. The predominantly reverberant acoustic field adds a further complication. In fact, Rainey³ has demonstrated that the reverberant sound field can dominate the direct pressure field even at the surface of the machine if the machine is very large. Other modifications to the near-field include proximity effects of adjoining components and sound reflections from the large machine surfaces which certainly occur in the case of the centrifugal chiller. Hodgson^{1,4} has referred to these effects, which would not be present for an individual component in a free-field environment, as "contamination" of the acoustic near-field around the machine.

NEAR-FIELD PRESSURE METHOD

For the case of the centrifugal chiller shown in figure 1 the structural vibration is primarily due to refrigerant flow fluctuations through the compressor. The sound pressure spectrum taken at a point 1.5 m above the concrete workspace floor and 1 m away from the compressor facing the machine is shown in figure 2. The noise is wideband and the spectrum level and shape is representative of measurements taken at any reference parallelepiped position as described by the ARI 575 standard for measuring machinery noise in equipment rooms.

An approximate ranking of the noise sources by sound power level was made using near-field 'A'-weighted pressure levels taken 2.5 cm distance from the machine surfaces as performed by Ebbing² and later by Hodgson.¹ In reference 1 it was shown that the method may only be regarded as a very rough estimate in that unrealistically high radiation efficiencies were obtained for the noisiest components (see Table I), namely the refrigerant discharge pipe and the cylindrical condenser, components numbered 10 and 11, respectively, in figure 1. However, the results clearly showed that the near-field pressure on the compressor side of the machine had contributions from both the condenser unit and discharge pipe. The condenser and discharge pipe appear to be nearly equal in sound power level. The unrealistic radiation efficiencies which have been calculated, particularly the value of 51.3 for the compressor side of the condenser, imply that the direct field component of the pressure is at least 17 dB below the total value measured (assuming the radiation efficiency was actually of order unity in magnitude). This means

that a more sophisticated near-field measurement technique is required in order to more accurately estimate the condenser and discharge pipe levels and hence the overall power level. Some care must be taken in the choice of the measurement method in order to cope with the detection of a signal at least 17 dB below the overall level.

SURFACE INTENSITY METHOD

Two known sophisticated source location techniques use the cross-correlation and coherence functions. For reverberant spaces, limitations of the first method are discussed in reference 1 while Halverson and Bendat⁵ have described data processing difficulties of the second method due to the long data sampling times required.

Another technique which has been mentioned in the acoustics literature but which has not been considered in detail for sound power measurement and noise source location is to make use of the exact result for the radiated sound power W , namely

$$W = \int \overline{p u_n} dS \quad (1)$$

where p is the fluctuating pressure at the machine surface having area S and surface velocity u . The time average $\overline{p u_n}$ is, of course, the surface acoustic intensity I_n and is normal to the surface.

INSTRUMENTATION

Measurements of the surface acoustic intensity I_n were made with a BK4344 2 gm accelerometer epoxied to the machine surface and a BK4136 0.66 cm dia. microphone placed in close proximity to the accelerometer and 1 mm from the machine surface, see figure 3. The frequency range of this transducer combination was evaluated by Hodgson¹ and extends to at least 8 kHz. In order to obtain $I_n = \overline{p u_n}$, an analog multiplier and averager may be used or the two transducers' signals may be recorded for convenience and digitally processed as was the case here. A 20-point survey and a 9-point survey were taken over the condenser and discharge pipe surfaces, respectively, with 2 or 3-point surveys for the remaining less noisy machine components. A typical tabulation of a survey, in this case for the discharge pipe, is given with a calculation of the 'A'-weighted power level in Table II.

Table I
Estimated Sound Power and Radiation Efficiency for
Cylinder and Pipe Based on Near-Field Pressure Data

Machine Component	Avg. Surf. Press. Level (dBA)	S Surf. Area (m ²)	Estimated Sound Power Level (dB)	Estimated Radiation Efficiency %
Condenser, rear side	102	10.17	112	7.6
Condenser, comp. side	106	10.17	116.5	51.3
Pipe	108.5	4.79	115.3	2.46

SOUND POWER LEVELS

The measured 'A'-weighted sound power levels for each component using the surface intensity method are shown below in Table III together with the calculated radiation efficiencies σ for the condenser and discharge pipe. The measured value of radiation efficiency is close to unity for the condenser and is typical of thick steel plates radiating with flexural wave motion close to or above the coincidence condition, see reference 6. Because of the greater surface area of the condenser, approximately five times the discharge pipe area, it was difficult to determine which component was the dominant noise source using the near-field pressure method. However, the results of Table III clearly show that the 'A'-weighted power level of the discharge pipe is, in fact, 8 dB higher than the compressor side of the condenser and 3.1 dB higher than the sound power radiated by the total condenser area. The overall 'A'-weighted power level for the machine is shown to be 109.4 dB for the particular chiller conditions and is dominated by the discharge pipe and condenser radiation. The 'A'-weighted power level as a function of frequency is shown in figure 4 for these two components. It can be seen that the discharge pipe is of order 15 dB higher in power level in the 2-3 kHz frequency range.

RESEARCH EFFORT AND FUTURE WORK

At the moment there is no other absolute sound power level data available for comparison with the results presented in Table III. However, the realistic acoustic radiation efficiencies which were obtained for the condenser and discharge pipe suggests that, at worst, the method is in error by only 3-4 dB. The research effort to date has consisted of one graduate student up to the completion of the Ph.D. dissertation, industrial support for the author for one summer, together with a research equipment grant from the New York State Science and Technology Foundation.

Further effort is needed to perform a series of accurate experiments on large components of various shapes and plate thicknesses radiating into different acoustical environments in order to establish the absolute accuracy of the method and also to determine the minimum number of survey points for a given accuracy. It is suspected that for the important frequency range of interest the surface motion is incoherent at relatively large separations. Figure 5 shows the coherence between two accelerometers mounted on the condenser surface as a function of frequency for two separation distances. When the accelerometers are closely spaced (25 mm)

Table II
Measured Values of Surface Acoustic Intensity
for Discharge Pipe ('A'-Weighted)

Survey Point	Rms pressure N/m^2	Rms velocity $m/sec \times 10^{-3}$	$I_n = \overline{p u_n}$ $watts/m^2 \times 10^{-3}$
a	3.9	6.0	12.4
b	3.5	5.8	10.6
c	3.4	7.4	12.4
d	3.1	5.8	9.5
e	3.5	6.9	15.1
f	3.3	6.3	13.9
g	3.5	6.1	10.9
h	4.4	6.1	14.4
i	3.9	5.5	9.8
Total			109

$$\text{Sound power} = 0.109 \text{ watts/m}^2 \times (4.79/9) \text{m}^2$$

$$0.058 \text{ watts (107.6 dB power level)}$$

Table III
Measured 'A'-Weighted Sound Power
Levels and Radiation Efficiencies for Chiller

Machine Component	Sound Power Level dB	Radiation Efficiency σ
Condenser, rearside	102.8	0.9
Condenser, comp. side	99.6	1.05
Discharge Pipe	107.6	0.42
Compressor	88	
Comp. Bearing	85	
Turbine	83.4	
Turb. Bearing	74	
Overall Power Level	109.4 dB	

the coherence is nearly unity so that the surface motion is coherent. But the coherence decreases to almost zero at large separation distance between the accelerometers (150 cm). This implies that areas of the condenser may be treated as incoherent sound sources so that simple summation of a relatively small number of surface acoustic intensity measurements is correct for overall power determination. Also, there is a need for a rugged, easily attached, microphone and accelerometer transducer combination with a portable readout device which would read surface acoustic intensity directly.

The current research program could be accelerated by adequate research funding. The present effort is limited to one master's thesis student. Since it has been established that the technique is feasible, further analysis of the nature of the incoherency of the surface motion is being performed. However, absolute comparison tests are sorely needed.

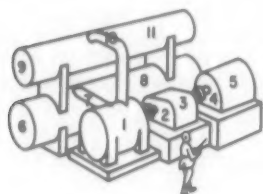


Fig. 1 Schematic of Large Centrifugal Chiller Showing Components

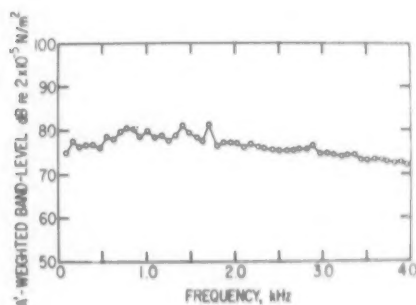


Fig. 2 'A'-Weighted Pressure Band-Levels for Chiller Machine Noise

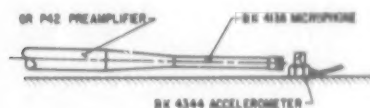


Fig. 3 Transducer Arrangement for Surface Intensity Measurements

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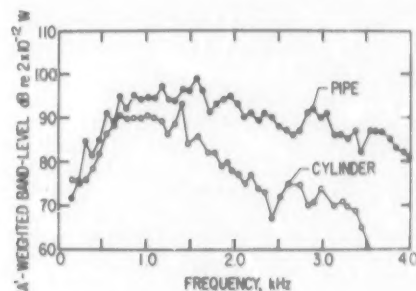


Fig. 4 'A'-Weighted Sound Power Band-Levels for Discharge Pipe and Condenser

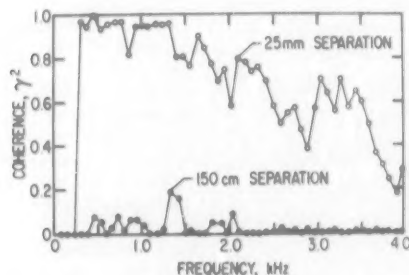


Fig. 5 Coherence Function for Surface Velocity on Condenser at Two Separations

"Sound Power Measurement
and Noise Source Location of Large Machines in Situ"
presented by
T. H. Hodgson

DISCUSSION:

T. Hodgson:

There was some grant support for this project from the New York State Technology and Science Foundation.

J. Tichy:

I think the method is excellent; it goes to the basics of the physics of the radiation of sound, and I hope it can be developed into a useable method.

R. Lyon:

A student of mine is also looking at this method. For calibration purposes, he has looked at radiation from a piston, and also from a simply supported plate both below and through the critical frequency. It turns out that if you pay attention to phase, you can get very good accuracy, indeed. The number of points required, as you mentioned, depends on the variations of intensity over the surface and particularly below the critical frequency you can easily have areas which absorb as well as radiates sound, so that you have to subtract as well as add as you go over the surface. This tends to be more of a problem on flat, thin structures such as machine housings or covers. I would like to ask you a question on the intensity. Why do you emphasize that there can be no tangential component of intensity at the boundary of the plate? Because of the stick condition which you introduce through vorticity, the tangential velocity of the air at the plate surface must be zero. I see no inconsistency in your statement.

T. Hodgson:

Perhaps I've belabored the point a little because some of the main criticism I've received on this method has been the question: How can the plate in flexure radiate tangentially when the motion is perpendicular to the plate?

R. Cohen:

I'd like to make a comment: In general many of these papers indicate a strong need for identification of noise sources. We don't have at this time, very good techniques. Harvey Nozick, if you and others are really interested at getting at the meat of a problem that would be beneficial across the board, it would be to develop a source identification technique that would be easy to apply and be economical.

O. McDaniel:

There is a technique called partial coherence function that purports to be the answer to all the problems. Apparently it takes into account additional sources that would contribute to the ordinary coherence function and subtracts them out.

T. Hodgson:

A couple of years ago, Chuck Ebbing and I tried to make this look simple to the practicing noise control engineer as to what these signal processing methods could do. I think the academic view of source identification is probably 5 to 10 years ahead of what industry is looking at at the moment. However, I would issue a warning: even though there are great benefits to these methods, there are great problems of interpretation. There has just got to be more effort put into them and to come up with case history studies to show that they are useful. You know you just can't make a blanket statement that says that partial coherence solves all identification problems, just as you could never say that correlation solved all jet noise source identification problems. I feel that there is a great opportunity for academia to get in and demonstrate by long term case history studies. I think there will be a different technique for each type of machine.

O. McDaniel:

Well, Bendat and Piersol conducted a seminar recently out on the West Coast on this technique. Apparently all it requires is some preprocessing before you go into the FFT, and they claim all sorts of wonderful things for it.

T. Hodgson:

I would just make one final remark that, as in all computing work, the intelligence is in the mind of the user not in the methods!

PROJECTS ON IMPROVING THE REVERBERATION TECHNIQUES
OF THE MEASUREMENT OF NOISE EMITTED BY MACHINES

Jiri Tichy, Professor

The Pennsylvania State University
University Park, PA 16802

INTRODUCTION

With the increasing emphasis which is being placed on the quality of our environment, and with the growth in the use of machines and mechanical equipment in industry, offices, homes and communities there becomes a definite need to characterize the noise emitted by these various noise sources in such a way that the data obtained from measurements at different laboratories, plants or measurement sites will be comparable.

Basically there are two methods used in describing the noise emitted by machines. 1. The sound pressure levels are determined either at one or more characteristic points at a specified distance from the machine. 2. The total sound power emitted by the machine is determined.

The method mentioned under 1 is normally relatively simple and quick, the major advantage being that the sound pressure level can be determined at the position of the operator and thus provide the data needed for his hearing protection. Also, this method can be useful to compare the same products on the production line, provided that the radiation characteristics of the product as a function of angle of radiation remains constant. However, for the design purposes of the environment in which the machine is operating, the information on the sound pressure level at one point at a given distance is meaningless.

To evaluate and characterize completely the noise of a machine, the information on the total radiated sound power has to be available in terms of the one-third or, at least, one octave band spectra. Knowledge of the radiated sound power is needed for at least two purposes: 1. To provide the designer of the machine environment with the data necessary to reduce the noise levels in the far field of the machine. This is particularly essential, if the machine is located in a room, shop or similar kind of enclosure. 2. For complete and meaningful comparison of identical or similar products (either manufactured by various companies or similar products but with different performance parameters manufactured by one company).

For the above-mentioned reasons, in the recent

decade a great deal of attention has been devoted to improving the precision of the methods for sound power measurements, so that industry can be provided with sufficient guidelines to make the measurements with sufficient reliability and at reasonable cost.

The sound power radiated by a source can be measured by two methods. 1. According to the first method, the intensity of sound, I , is measured on a sphere or other suitable hypothetical surfaces around the machine and the total power is determined by

$$W = \iint_S \vec{I} \cdot d\vec{s}. \quad (1)$$

When making practical measurements, the integration is replaced by the summation over a sufficient number of points in which the sound pressure is measured. Also, it is assumed that the intensity can be calculated from the relation $I = p^2/\rho c$, and that the direction of $\vec{I}$ is identical with the normal to the surface ΔS . Methods with a suggested number of points and their recommended locations have been developed and the probable errors were estimated. This method is generally applicable to large size sources which cannot be placed in a reverberation room. In order to reduce the error due to the environment, the measurement should be performed either in an anechoic chamber (which may have a hard floor if this is a typical source installation) or in a semianechoic environment.

2. This paper deals with the second method for sound power measurements, which employs the reverberation room as the measurement environment. Evidently, by its nature, this method is limited to small or medium size sources but offers advantages consisting of fairly good precision in the results as well as reasonable cost of the facilities and quick measurement procedure, particularly if modern electronic equipment for signal processing is being used. For this reason both industry and research or testing laboratories are employing this method as the major method for sound power determination.

This paper presents the results of the research in

the reverberation method for sound power measurements performed at the Department of Architectural Engineering of The Pennsylvania State University. The research has been sponsored by the grants or contracts with the following agencies or companies: American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc., B & K Instruments, National Science Foundation (2 grants) and National Bureau of Standards.

PRINCIPLES AND PROBLEMS OF SOUND POWER MEASUREMENTS IN REVERBERATION ROOMS

The determination of sound power in reverberation rooms is based on the assumption that the reverberant sound field has a uniform energy density and that the energy density is proportional to the power radiated by the source. It is often also assumed that the sound pressure squared is proportional to the energy density, but this is true only if the pressure squared is averaged over the entire volume of the reverberation room.

The practical procedures to determine the sound power in the reverberation room are described in the ANSI S1.21 "Methods for the Determination of Sound Power Levels of Small Sources in Reverberation Rooms". This standard defines the limits of the precision of the sound power determination, which can be expected if all provisions of the standard are satisfied and, particularly, if the chamber is qualified according to described qualification procedure for a particular type of source. Generally, the precision of the measurements depends on the source spectrum. If the sound source radiates a broadband spectrum, its power can be measured more accurately than the power of a source that radiates a narrow band spectrum or single frequencies [1]. The research sponsored by the previously mentioned agencies concentrates mainly on the problems of measurements at single frequencies because of the general difficulties involved.

There are two major sources of error in determination of the sound power: 1. The variations of the actually radiated sound power with the location of the source in the room and 2. the error introduced by measuring the sound field at a limited number of points. Additional error can be caused by the lack of precision of the equipment for signal processing, microphone calibration and the like, but these errors can be usually kept well under control by the room designer or operator.

The variations of both the sound power radiation with the source position and frequency, and the sound pressure spatially in the room are best described by the variance of the statistical laws for these variations.

The variations of sound power with source position p^2 have been predicted theoretically [2]; however, the experimentally determined values are much lower, particularly at low frequencies [3]. Additionally, the sound power averaged, for instance, over a band of frequencies that is radiated into the reverberation room, is lower than the sound power which the same source would radiate into free space [4], particularly at low frequencies. The

reverberation room method has been frequently criticized for this systematic error. However, it must be considered that very few sources radiate sound into a free space, and therefore, the results of measurements in a reverberation room can characterize the source more realistically.

Another source of error in the determination of the sound power is an insufficient sampling of the interference pattern of the reverberant portion of the sound field [5], particularly if the source radiates pure tones, so that the maxima and minima of the sound pressure have considerable level difference. The effectiveness of the sound pressure squared averaging depends on both the spatial variability and the number of uncorrelated samples taken. At high frequencies, when the sound field is sufficiently diffuse, the variance of the average p^2 obtained from N samples to the true mean p^2 in the diffuse field is given according to Schroeder [6] and Lubman [7] by

$$\sigma_p^2 = \frac{1}{N} \quad (2)$$

At frequencies lower than approximately $f = 2000 \sqrt{T/V}$, where T is the room reverberation time and V its volume, the variance is given by

$$\sigma_p^2 = \frac{\sigma_0^2}{N} \quad (3)$$

where $\sigma_0^2 > 1$. Although no theoretical prediction for σ_0^2 has been obtained, experimental data are available [8].

When making actual sound power measurements, the total variance, σ_t^2 , of the sound power determined in the reverberation room is given by the sum of the two variances:

$$\sigma_t^2 = \sigma_w^2 + \sigma_p^2 \quad (4)$$

To determine the suitability of the reverberation room for the sound power measurements, the room has to be qualified according to the procedures described in the ANSI S1.21. This procedure does not determine σ_t^2 directly, but involves the uncertainty of the sound power determinations based on both effects described previously.

The basic idea of the qualification procedure is to test the chamber under the same conditions of its actual use. The sound source for the qualification test is a small loudspeaker (representing a point source) that is placed in the position of actual sources tested. The sound pressure is sampled and processed with the equipment designed for that particular chamber. The chamber is tested for each one-third octave band of the required frequency range of the chamber. The particular one-third octave band is subdivided regularly into 22-25 frequency intervals and the loudspeaker response in the free field is determined at these frequencies. The loudspeaker is then placed in the chamber and the spatially averaged value of the sound pressure level is determined for each of the 22 frequencies. After correcting these values for the loudspeaker calibration, the standard deviation of the sound

pressure level value is calculated. If the magnitude of this standard deviation exceeds the value defined by the standard, the chamber does not qualify for measurements.

It is apparent that the qualification procedure involves the combined effects of the frequency dependence of the sound radiation and the p^2 averaging. In order to find out to what extent the existing reverberation rooms qualify for measurements at pure tones, many chambers were tested on an international basis [9]. It was found that none of the classical chambers with hard walls qualified at low frequencies.

What are the means to achieve qualification and increase the precision of the measurements? The sound pressure averaging was improved by the use of a rotating diffuser, which introduces amplitude and frequency modulation of the sound field and reduces the variance σ^2 under the value of $1/N$. With a suitable diffuser design a considerable gain can be achieved.

The variation of the radiated sound power with both frequency and position can be decreased by increasing the modal damping and thus improving the modal overlap. Low frequency absorbers placed on reverberation room walls represent a practical solution to the problem.

The research sponsored by the above-mentioned agencies contributed some data for the design of reverberation rooms which allows measurement of the sound power with better precision. The results of this research are summarized in the next section.

RESULTS OF RESEARCH

The research conducted during the 5 year period at The Pennsylvania State University was oriented in two directions. 1. To explore the basic knowledge of the statistical properties of the sound fields, modulated by the moving diffusers, and the effect of increased modal damping on the properties of the sound field in the reverberation room with or without the moving diffuser. Also, an extensive study on the effect of source position on the sound power radiation has been conducted. 2. Another aspect of the work was related to the application of the gained theoretical and experimental knowledge on the guidelines for design or reconstruction of existing rooms, so that the measurement of sound power could be done with better precision. Although many problems have been solved, there are still many open questions and further research will be necessary to find the answers. It has to be recognized, that particularly the research on the effects of rotating diffusers is from the theoretical point of view a completely new area. The effects of the time varying boundary conditions on the sound field is not covered by any publications and therefore the fundamental principles of the approach had to be found. Also, the experimental research programs related to gathering practical data for direct applications was based on measurements of a statistical nature, which require advanced electronic equipment and the collection of a great amount of data.

This paper does not present any detailed material on the research results but only briefly summarizes the achievements with references to literature containing the details.

1. Research Related to the Effect of Rotating Diffusers

The first task was to determine the basic changes in the sound field, generated at a single frequency when a rotating diffuser was applied. From measurements in an anechoic chamber it has been found, as expected, that the moving diffuser (when reflecting the sound waves because of the Doppler effect involved) is actually a frequency modulator. If the incident wave has a single frequency, the spectrum of the reflected wave consists of several lines, with frequency intervals dependant on the speed of the diffuser, centered around the incident wave frequency. Using the frequency modulation theory for electrical signals, the modulation index of simple shaped diffuser could be determined. It was found that the larger the diffuser and the higher the modulated frequency, the broader could be the spectrum of the reflected wave. When operating the rotating diffuser in a reverberation room, the spectrum becomes even broader by reflections of sound from the walls. This is due to each spectrum line which when reflected from the diffuser generates a new line spectrum. Additionally the diffuser modulates the amplitude of the sound pressure in the room. It could be concluded that the final effect of the rotating diffuser in the sound field is similar to the generation of the sound field by a multitonal source and; therefore, the spatial variance of p^2 in such a sound field is much smaller than when the source radiates a single frequency. For practical purposes, the reduction of the variance can be described in terms of a Figure of Merit. This means that the number of microphones used for the scanning of the sound field can be either reduced or, with the same number of microphones, the precision of the results is improved [1, 8]. Although several kinds of diffusers were tested, more work has to be done before any conclusion on an optimum design and placement of the diffuser in the reverberation room can be made.

Another study of the diffuser was directly related to the qualification procedure of the room as defined in ANSI S1.21. This study was performed in combination with the damping of the modes, and the results were expressed in terms of the wall absorption coefficient [8]. It was found that with highly absorbing walls and without the diffuser operating, the room does not qualify. This was in agreement with the tests performed on an international basis. Also the possibility of qualifying the chamber depends on the source position. When operating the diffuser the qualification criterion was approached, but at low frequencies in particular it has been found essential to increase the wall absorption of the room to achieve the qualification. For a so far unknown reason, the increased chamber wall absorption ($\alpha \sim 0.16$) at middle and high frequencies increases the variance of the sound pressure and the room is less likely to qualify than with low wall absorption ($\alpha \sim 0.05$). Therefore, we have concluded that from a practical point of view only

the increase of the low frequency absorption can be recommended and if the qualification has to be achieved, it is essential to increase the modal damping in the low frequency region.

Qualification for multiple source positions was also experimentally investigated and the results have been found to correspond with the recommendations in the ANS S1.21. Because the multiple source position method is not too practical, this problem was not deeply explored.

Another problem related to the diffuser operation which was pursued was the extent of the diffuser effect on the sound power radiation from the source. There were indications [10] that the instantaneous power radiated by a monopole source is a function of the instantaneous diffuser position and; therefore, it was hoped that the diffuser will also reduce the spatial variations of the radiated sound power as the source is moved throughout the chamber. However by careful measurements [3] it has been found, that the diffuser has negligible effect on the variations of the radiated sound power with source positions.

The radiation of sound from the source during one diffuser revolution will have to be studied further. A special source allowing the instantaneous measurement of radiated sound power will have to be constructed, which as a preliminary study indicates, is quite a complex task.

2. Theoretical Study of the Time Varying Boundary Conditions

In order to base the experimental results on the theory, a theoretical model of the reverberation room as been developed [11]. To simulate the time-varying boundary conditions a rectangular room has been considered with one wall oscillating at a large amplitude (up to one wavelength of the sound source frequency) and at a low frequency corresponding to the relatively slow speed of the rotating diffuser. The mathematical treatment was extremely complex, however the model allows the calculation of the sound pressure in the room at any point and for any any monopole source position with the frequency and amplitude of the oscillating wall as a parameter. Another parameter is the impedance of room walls.

Although the mathematical solution was basically obtained in a closed form, the actual determination of some coefficients is based on approximate solutions of an infinite number of equations. Also, in order to evaluate the statistical parameters such as the spatial variance of p^2 , and the actually radiated power by the source requires the use of computer programs which entail several hours of computation on a fast computer (IBM 370) and very large memory (at least 300K).

Using the developed theory and computer programs a series of calculations has been performed. Because of time and financial limitations, only a limited variety of parameters have been studied, and it is hoped that further financial support will be found to continue in this work. The correspondence with the measured data on rotating diffusers has been found very satisfactory. The results are summarized

in the National Bureau of Standards Report [12].

First, the effect of wall amplitude was studied. It has been found, that the spatial variance of p^2 decreases with increasing amplitude of the wall oscillations, while the frequency of the wall oscillations had a very small effect on the variance of p^2 . This result is quite logical, because the increased amplitude is causing deeper modulation of the sound field. The amplitude of the wall oscillations is analogous to the size of the rotating diffuser.

Another group of calculations were devoted to the study of the effect of wall absorption on the spatial variance of p^2 . Calculations were performed for 22 frequencies of the 125Hz one-third octave band with the boundary fixed and with the wall oscillating at a constant amplitude. The values of the spatial variance were different for each of the 22 frequencies. However the averages over the one-third octave band indicate that with increasing wall absorption (from $\alpha = 0.04$ to 0.12) the averages of variance are decreasing from 1.58 to 0.75 for the wall oscillating. The calculations were performed for two source positions: one chosen randomly inside of the room and one very close to the wall to simulate the two probable source positions. In both cases the effect of absorption was similar and also, the Figure of Merits were not too dependant on the source positions. It can be concluded, that at low frequencies the increased modal damping gives the advantage of decreased variance of p^2 . If this is also true, higher frequencies will have to be found.

Another series of calculations were devoted to the study of the effect of the wall absorption and the wall oscillations on the actual power radiated by a monopole source into the reverberation room. In this case, 10 source positions were chosen: 5 inside of the chamber and 5 very close to the boundary.

The calculations averaged over 22 frequencies of the 125Hz, one-third octave band indicate that the power radiated into the chamber with the fixed boundary is systematically less than the power which the source would be radiating into free space. The difference for a particular source position is as much as 10 dB. The power averaged over 5 source positions results in the difference of 3 to 4 dB depending on the absorption. Increased absorption decreases this systematic error slightly (by approximately 1 dB). The effect of wall oscillations on the radiated sound power is very small. This confirms the experimental findings with the rotating diffuser [3].

Finally, the calculations of the qualification procedure have been performed. As expected from experimental results on rotating diffusers, increased modal damping and the wall oscillations are essential to achieve the qualification.

As mentioned before, the description of the results represents only the major line of the research. More detailed work has been done.

SUGGESTIONS FOR FUTURE WORK

Although many experimental data as well as data from the computer model have been found, there is a great need for further continuation of the research.

To better understand the effect of a rotating diffuser on the sound field, the modulation effects of the diffuser will have to be studied in more detail. Another study should be oriented to find the most efficient shape and position of the diffuser in the reverberation room.

Another important area of research is the radiation of sound power into the chamber. So far only small monopole sources have been considered. Actual sound sources are much more complex and the future work should involve dipole and higher order sources. Also the effect of a rotating diffuser on the instantaneous power radiated from a source should be further explored.

The analytical model should be applied to other frequency ranges besides the 125Hz, one-third octave band, particularly in view of the very good results obtained so far with this model.

Another area to be explored is the statistical distributions of p^2 with the rotating diffuser. Both theoretical and experimental studies should be conducted. Although a theory for the statistics of power radiation into the room has been suggested [2], the difference between the actually measured values suggest the need for further theoretical work in this area. Also, neither a theoretical nor an experimental study has yet been conducted of the statistical functions of the power radiated into the reverberation room as a function of frequency.

These are only basic suggestions and it can be expected that future work will open new problems for the research.

Because all this work is related to the precision of the measurement of the sound power emitted by various noise sources and the knowledge of the sound power is essential to both their comparison and rating, it is believed that sufficient financial support will be found to continue in this research.

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"Projects on Improving the Reverberation Techniques
of the Measurement of Noise Emitted by Machines"
presented by
J. Tichy

DISCUSSION:

P. Beade:

ASHRAE Standard 36-72 which you had listed as the other standard on sound power measurements in reverberation room has just been withdrawn. The reason is that now ANSI S1.21 is well established and we feel there is no more need to have a separate standard. The other point that I would like to make is that amongst the various measurement methods, the reverberation room method may appear to be unduly complicated because the standard which covers it is "fatter" and there are more "does" and "don'ts" than on the other methods. However, I would caution that this is not a proper conclusion. The problem is that the other methods have not been brought up to date for the last 15 years. They are still only covered by a totally obsolete American Standard S1.2. The revision of the other methods is long past overdue, and the fact that they have not been brought up to date constitutes, in my estimation, a very serious gap in our national measurement system that I hope the National Bureau of Standards will recognize.

R. Cohen:

It appears to me that in a broad-brush point of view, that reverberation chambers used for sound power measurements are not found in universities except for people who are doing testing, or who are doing research into finding better ways to make measurements in that type of chamber. On the other hand, you find many of these in industry because of the simplicity of the measurement. It is my opinion that the people who have these chambers, unless they are lucky enough to have a good acoustical engineer, don't know how to use them. Can we tell industry that if they make the measurements in these rooms that they can expect results within a certain accuracy?

J. Tichy:

Yes, I think we can tell them that. The error between free-field and chamber would be about 3 to 5 dB, on the average.

A NOISE CONTROL PROJECT IN A SNACK-FOOD PROCESSING PLANT

D. Muster, Brown and Root Professor
Department of Mechanical Engineering
University of Houston
Houston, Texas 77004

ABSTRACT

In food-processing plants there are unique noise control problems. On the one hand, federal guide lines require that the plants be quiet. On the other, regulations governing cleanliness require frequent washdowns of walls and equipment. Thus, plant engineers seek soft (for noise) - hard (for cleanliness) solutions to their problems, an almost self-contradictory set of requirements.

In the paper, the general nature of the functional requirements for the equipment in a snack-food processing plant is discussed. The results of an initial survey which defines the noise control problem are given. The sources of noise include conveyors, air jets, combustors, extruding and packaging machines, air-moving equipment and large panels of stainless steel in vats, tanks and ovens. The approach selected to solve the problems combined an educational program with the development of practical solutions to the worst problems. Concurrently, in concert with the corporate engineering staff and managers, a long-range plan for noise-control was developed. The plan provided for the issuing of engineering instructions to plant managers and engineers, developing an improved equipment maintenance schedule, the inclusion of low-noise requirements in new-equipment procurement specifications and a continuing program of education and training in noise-control technology for plant personnel.

INTRODUCTION

There is a unique character to the sometimes-contradictory requirements placed on the designers and operators of snack-food processing plants. This suggests that it would be desirable to set the context of their noise control problem before proceeding. Snack foods, as we know them, are a relatively recent innovation, except for popcorn which has been an American foodstuff at least since the time of the ancient deposits

in Bat Cave, which were in place 4000 years ago¹. In their modern forms, we have popcorn, potato, corn and wheaten chips, wheaten pretzels and crackers, and bits of cornmeal extruded at high temperature and pressure so that they puff into a variety of forms for subsequent drying and, perhaps, coating with oil or other ingredients.

The surfaces of the equipment used in snack-food processing plants are preferably stainless steel; no copper, bronze, brass, Monel metal or other copper alloy is permitted to come in contact with the fats with which the bits and pieces of food are fried or coated. If cleanliness were the only consideration, any of the copper alloys above or stainless steel would suffice. However, the oil layer of the fried snack is the source of a serious problem. The limit to the shelf life of snack-foods is determined primarily by the rancid taste which develops from the slow oxidation of double carbon bonds in the oils. The reaction is catalyzed by light and trace amounts of metal ions, particularly those of copper. Hence the requirements that only stainless steel be used wherever fats or fat-covered food comes in contact with the equipment and that light-imperious packages be used.

The equipment includes specific machines for performing the following functions (usually performed in the order given):

- receiving the raw foodstuff (potatoes, corn and wheat) in up-to freight-car quantities;
- cleaning and preparing the raw foodstuff (say, removing the eyes of the potatoes, soaking and shaping the cornmeal mix, etc);
- slicing or extruding;
- frying;

¹Matz, S. A., *Snack Food Technology*, Avi Publishing Company, Westport, Connecticut, 1976.

- drying or baking; perhaps
- coating the product with oil, salt, spices and/or cheese;
- packaging; and always
- conveying the product to, from and through the equipment.

The processing is completely automatic, mechanized from receiving the raw food-stuff to packaging. Matz describes the engineering and design of typical machines; equipment is described clearly with many references to the specific equipment available¹.

As an example, let us consider the sequential operations and equipment necessary to make a corn-puff product at a typical rate of 300 pounds of cheese curls per hour. The cornmeal mix is soaked in large (10 ft. x 2 ft. x 2 ft.), wheeled, stainless steel vats. After the bulk of the fluid is removed, the damp cornmeal mix is formed into 18-inch long, 6-inch diameter cylinders. These are fed into pressurized extruding machines with a water-cooled die head which produces the puffs. A conveyor passes the puffs through a drying oven with a large flow-through capacity for hot air. Another conveyor carries the dried curls to a tumbler where (while still hot) they are coated with oil and a spray-dried emulsion of cheddar in powder form. From the tumbler the finished puffs are transported by a long (vibrating) conveyor to the packaging machines. The conveyor is long in order to let the curls cool. At each machine the off-loading from the conveyor is accomplished by high-pressure air jets which sweep the curls off the conveyor and into the maw of the packaging machine. The latter temporarily stores, meters (by weight) and packages the curls, usually in bags which are made by the machine from rolls of imprinted plastic film. Here, man enters the scene to pack the packages in shipping cartons. Auxiliary equipment for preparing the mix and other not-on-line functions includes blenders, mixers, pumps and blowers. Usually, the plant is equipped with a large-capacity air-moving system to remove the hot, moist, grease-laden vapors to the outside.

Because of their short shelf life, snack-foods are usually sold within relatively short distances (say, 100 miles) of the processing plant. One of the large companies operates over 50 plants. Usually each one produces the same basic product line. Thus, each plant is about the same size and contains the same equipment.

DISCUSSION

The layout of the plant at which the project here was conducted is shown in Figure 1. Its configuration, equipment arrangement and size are typical of many similar

plants. In any shift the work force normally in the processing areas is approximately 45 men and women. Administrative personnel do not normally enter the processing areas.

For easy cleaning, the walls, floors and ceilings of the processing rooms are tile or other hard surfaces (such as finished cement). The fryers and drying ovens are located in large rooms with hard-surfaced walls and floors, but the ceilings are relatively high (about 30 ft.) in order to provide space where the hot, moist air rising from the equipment may collect and then be removed through the roof by large (5-ft. diameter) fans.

There are work stations in the processing rooms at the positions shown in Figure 1. The individual tasks of the majority of the work force requires that they remain at their work stations or in one room only. However, those who maintain and service the equipment move throughout all parts of the processing rooms.

The enlightened management of the company was aware of the OSHA guidelines for acceptable levels of noise exposure in industrial environments. They wanted to know the company's position vis-a-vis these levels of exposure. In particular, they wanted to know if there were areas of potential noise hazard in their plants. If they existed, what might be done to reduce the noise levels to the acceptable limits. Finally, what plan could be proposed to make the control of noise in their plants a minimal problem in the future. At that time there were no engineers in the company (on the corporate engineering staff or in the plants) who had taken training in acoustics or noise control technology.

In this situation, the company decided to hire an outside consultant (a university professor) (1) to make an initial noise survey of the processing areas, (2) to analyze the results of the survey and define the company's problem, and (3) to present recommendations and a plan of action.

The survey results are summarized in Table 1. The noisiest sites were (1) in the packaging room underneath an elevated vibrating conveyor (95-98 dBA)² and at the side of the packaging machines (90-92 dBA); (2) at work stations near the combustors which heated the oil in the fryer vats (94-96 dBA); (3) near the containers into which the raw potatoes were first received

²Occasionally, the levels under the conveyor exceeded 100 dBA. It was found that in these cases the excessive noise was due to bad bearings in the out-of-balance motors used to generate the impact forces necessary for the operation of the conveyor.

(88-92 dBA): near the cornmeal mix extruders (89-90 dBA); and near the air jets used to sweep the product off a conveyor onto another conveyor or into a packaging machine (intermittent noise, to 92 dBA). The overall levels at even relatively quiet locations varied from 82-87 dBA. Thus, no area in the processing room was so far below the acceptable 90 dBA level as to not be of concern. There were several areas where the hazard was real (as defined by the OSHA guidelines).

The survey results were discussed with the engineer who had been assigned the task of organizing the company's effort to reduce noise in its plants. With his collaboration, the consultant prepared the following one-year plan. It was presented to the corporate management and approved. It was agreed that

- (1) the consultant would
 - (a) prepare and present a series of lectures on acoustics and noise control to selected engineers of the corporate engineering staff.
 - (b) assist in the establishment of a corporate-level acoustics laboratory. (This would be done by providing a schedule of noise-measuring and analysis equipment which would be purchased over a two-year period. A first-year budget of \$10,000 would be provided.)
 - (c) prepare (in consultation with the engineering staff) a long-range plan concerning the control of noise in the company's plants.
 - (d) provide an engineering consulting service to recommend methods, design changes and procedures for reducing noise at the locations where the noise levels were in excess of OSHA guidelines.
 - (e) provide the company engineering library with a list of suitable texts, journals and other technical literature.
- (2) the company would
 - (a) furnish funds in a one-year budget to support the items listed above.
 - (b) assign an adequate group of engineers and technicians to the noise-control project laboratory (Note: 6 engineers attended the class, 3 of whom were assigned permanently to a newly established noise-control section of the engineering staff.)
 - (c) provide adequate facilities for the laboratory and lectures cited above. The expectation was that at the end of the year the company engineers would be adequately trained and equipped to operate with only occasional consultation from outside the company.

In Table 1 the principal noise problems and sources are tabulated. The general characteristics of the noise sources are noted. An attempt was made to solve the worst problems immediately. The measurements, analysis and solutions were conducted with

the students as case studies in the course. The approaches to these problems are discussed below.

Packaging Room

There were two noise-hazard areas in the packaging room. One was at the rear of the packaging machines underneath an elevated vibrating conveyor. A second was immediately adjacent to the packaging machines at their sides. Air jets used to control the flow of the product off the conveyor contribute to the overall noise in the room.

• Conveyor noise. The conveyor is a smooth-surfaced, shallow, stainless steel pan. It is set on a slight angle along its length (40 ft.) so that the delivery points for the product are lower than the entrance point. The entire conveyor is hung from the ceiling so that it is about 8 feet from the floor. An out-of-balance electric motor is mounted directly to the bottom of the pan so that it delivers a periodically recurring force to the pan. The force causes the pan to vibrate and the product flows to the lower delivery points at the packaging machines.

Immediately below the conveyor there is a confined space (8 ft. x 3½ ft. in vertical cross section x 40 ft. long) with a hard-surfaced wall and floor on two sides, the conveyor bottom and backs of the packaging machines on the other two. Maintenance personnel and others regularly move through the space.

The sound spectrum here is dominated by the impact noise of the conveyor power source and the noise radiated from the bottom of the conveyor. The packaging machine contributes a lower-level collection of components due to the interaction, impact and vibration of its parts. The product (as it moves) contributes a lower-level, broad-band background noise, a shushing sound caused by the rutting of the salt-roughened surfaces of the myriad of chips on the conveyor. The combined level from all sources is from 95-98 dBA. In two instances the noise from the conveyor exceeded 100 dBA (102-104 dBA intermittently). It was a clattering sound caused by bad bearings in the electric motor driving the conveyor.

Two approaches were attempted to reduce the conveyor noise: (1) a damping material was applied in patches in several patterns to the bottom of a sample section of test conveyor. The reduction achieved was significant (approximately 9 dBA) under the test conditions; however, the application of a similar treatment to an actual conveyor would be difficult and expensive, although it would be relatively easy to apply

at the time the conveyor was fabricated. As a consequence, the final recommendation was directed at the long-range problem. All future acquisitions of similar conveyors (which are functionally excellent) would include a low-noise requirement as a part of their purchase specification. Further, when the existing conveyors are shut down for a long period, an attempt will be made to treat them with damping material.

The product noise remains unchanged and untreated; however, it is expected that damping the conveyor pan will probably reduce the noise from this source as well. The surfaces of the product (roughened by salt, etc.) must be accepted as is and preclude any serious attempt at reducing the noise caused by their rubbing. In addition, the rubbing noise is less objectionable than the noises from other sources.

The bearing noise problem was easily solved; the bearings were simply replaced. For the long-term, an improved maintenance and replacement schedule was instituted and, in future acquisitions of the conveyors, improved bearings will be requested.

In summary, it was recommended that the conveyor noise can be reduced by applying a damping treatment to the bottom surface of the conveyor pan, the bearing noise can be eliminated by replacing the bearings, and (for the present) nothing can be done about the product noise. In the future, the conveyor manufacturer will be required to reduce the noise of his product. Acoustic treatment of the space behind the packaging machines does not appear to be a feasible solution.

• Packaging machine noise. The sources of noise in the packaging machine were mechanical plus the acoustic noise from the exhaust air of pneumatic lever actuators (90-92 dBA). The noise sources in the machine are many. Unfortunately, detailed examination of the machine is not feasible; thus, no significant reduction in overall noise is possible. On this basis, it was agreed that (1) the exhaust posts would be redesigned so as to reduce the noise from this source and (2), in future installations, an acoustic barrier would be installed between the machines and their operators. An immediate reduction in overall noise at the work stations was brought about by a simple change in the acoustical character of the room. Large packages of folded corrugated-paper shipping containers were brought near the work stations. By storing them so as to cover a large portion of the rear wall, the overall noise in the room was reduced about 2 dBA, enough to make the work-station noise-levels only marginally hazardous.

In summary, the overall noise of the packaging machine is due to many sources only one of which (the exhaust noise from the pneumatic lever actuators) is amenable to immediate modification. An improvement in future installations will be possible by installing an acoustic barrier between the machine and its operator. By storing the packages of shipping cartons solidly against the rear wall of the room, the acoustic characteristics of the room are improved without interfering with the cleanliness requirements.

• Air jet noise. Air jets are installed at about 115° to the flow-line of the product along the conveyor. The jets sweep the product off the side of the conveyor onto another conveyor or into the storage bin of the packaging machine. The jets are simply valved pipes which use 100-psi shop air. Their noise is a major component of the 95-98 dBA measured beneath the conveyor pan. Near the jets it is slightly higher. Since quiet-flow jet devices are available, the recommendation was made to specify their use.

In summary, air jets perform well in changing the direction of flow of the product without damaging it. Thus, it was a better choice to select an improved (quieter) jet than to replace them with mechanical devices which would have greater potential for damaging the product.

• Fryer combustion noise. Four combustors are used in each fryer, each one inserted into the combustion chamber through an oversize circular opening in the wall of the fryer. At a nearby work station, the noise level was 94-96 dBA. For optimal combustion, the combustors require great volumes of air and turbulent mixing, a combination which also causes the high-level, roaring sound associated with it. After consultation with the manufacturer of the combustors it was recommended that (1) the air inlets be modified, a circular tube be used to encompass the combustor along approximately two-thirds of its length, (2) the fuel/air mixture be adjusted so as to minimize the noise. The broad-band, low-frequency noise of the combustion process precluded achieving a significant reduction by changing the air inlets from holes to tubes (there was only 1-2 dBA improvement). The greater reduction can be achieved by reducing the rate of combustion (so as to reduce the requirement for air) and adjusting the fuel/air mixture. It appears that a 4-5 dBA reduction will be possible by this means.

In summary, the only immediate reduction in fryer combustion noise could be achieved by reducing the heat demand, or by adjusting the fuel/air mixture and reducing the

total volume of air required. In the long-term, an improved airlet to the combustion chamber may provide an improvement.

• Other noise problems. Potatoes entering the production line in the receiving room hit the sloping sides of large tanks of rectangular cross-section. This causes the stainless steel panels of the tank to vibrate. The resulting low-frequency rumble was not always excessive. It could be reduced to below acceptable levels by stiffening the tank side panels and by providing a rubber, pad-like anvil for the potatoes to strike when they entered.

SUMMARY AND CONCLUSIONS

A survey made in a snack-food processing plant showed that there were several work

locations where noise was a potential or real hazard (as defined by the OSHA guidelines). A plan was developed to train company engineers in acoustics and noise-control technology, using the solutions for the worst problems as case studies. A long-term plan was developed with the collaboration of company engineers and managers which included the establishment of a noise-measurement laboratory. Engineering instructions were issued to correct existing noise problems. New or replacement equipment will be purchased with low-noise as a contract specification, where possible. The design of new plants will include features which will insure that noise levels will be below OSHA guidelines. An enlightened management and enthusiastic and receptive engineers are a delight to work with as a consultant.

Table 1
Noise Levels in Processing Areas
(Levels in dBA)

Function (Area)	Receiving	Preparation	Slicing	Extruding	Frying	Drying	Conveying	Packaging
Source		Ae 1		Ac 1	Ac 2	Ae 1	Ac 1	Ac 1
Product	M1*	M3	M3	M3	M2,3	M2,3,5	M2,3,4,5	M2,3
Corn chips, Corn puffs, Wheat chips	— o —**	85 dBA		89-90 89-92 for puffs	94-96	86-88	95-98***	90-92
Potato chips	88-92	— o —	87		94-96	86-88	95-98***	90-92

*Noise sources:

- Acoustic (Ac) 1. Air jet (piercing sound; broad-band, high-level with high-frequency components)
- 2. Combustion (roaring sound; broad-band, high-level, low-frequencies)
- Aerodynamic (Ae) 1. Air flow (general background noise; broad-band, intermediate level)
- Mechanical (M) 1. Panel vibration, impact (intermittent, low-frequency rumble)
- 2. Panel vibration, forced (multiple-frequency noise)
- 3. Machine parts (interaction, vibration and impact noise)
- 4. Machine part, bearing (narrow-band with strong single-frequency component)
- 5. Product rubbing (relatively loud shushing sound due to rubbing surfaces)

** No problem

*** See Footnote 2.

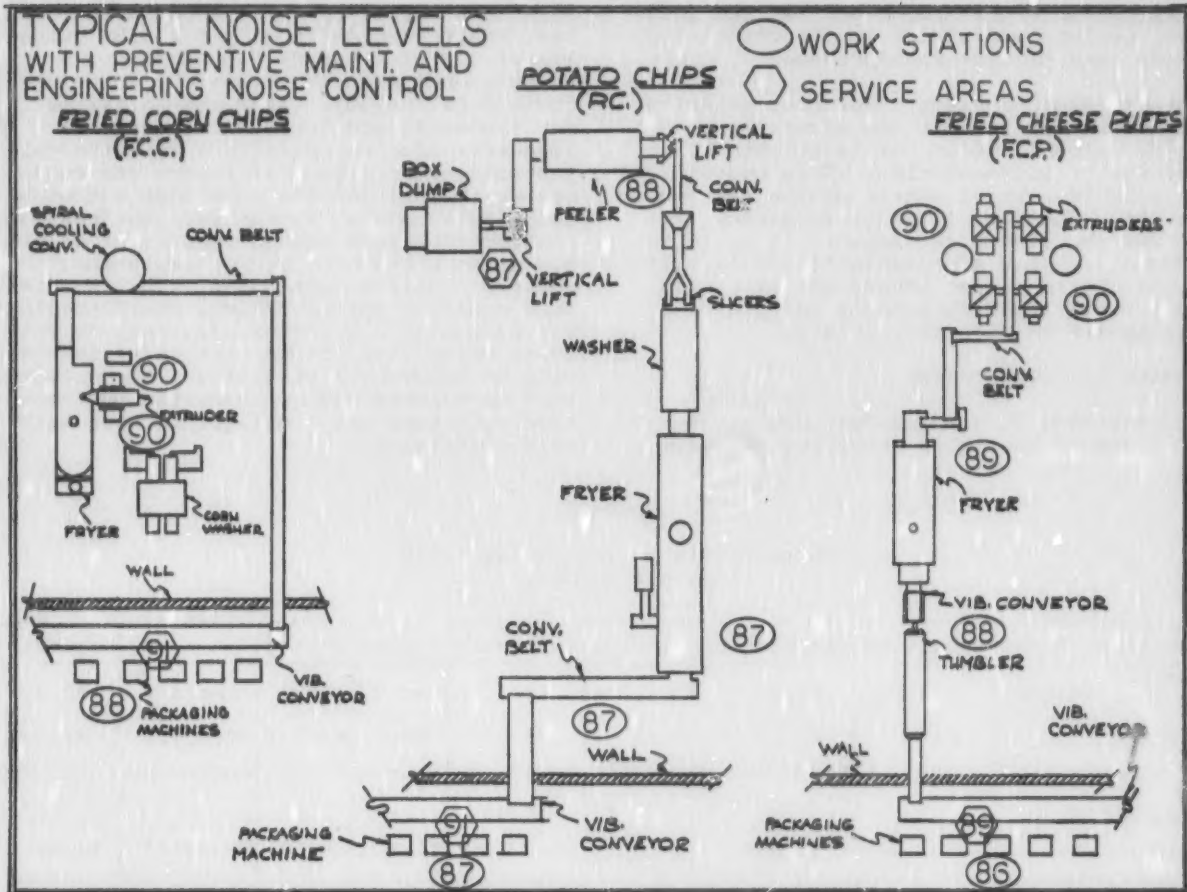


Figure 1

"A Noise Control Project in a Snack-Food Processing Plant"
presented by
D. Muster

DISCUSSION:

H. Nozick:

There are some cases where industry will do just enough to meet a requirement. That's fine. In the course of your investigations have they and you tried to get below the requirements?

D. Muster:

This is the third time that I have done this. Of the three, this is the one that's been most successful. I think what you are asking is a measure of the enlightenment of the management. Even where there is an enlightened management, the pressures don't permit them to go much below. On this basis, if I were part of their management, I would not go below either.

H. Nozick:

I understand exactly what you're saying, but have they done any cost-benefit studies, for example, to determine if they should go below for their own personal benefits?

D. Muster:

I would say yes, to the extent that one of the first things I asked them to do was to look at their maintenance program. I've found that if you have a good maintenance program, you'll not have noise problems to the degree that you will if you have a poor program. Other than this, they did not consider other cost benefits.

H. Nozick:

The levels that they reduced to, 87-90 dBA roughly, can still be harmful to the employees. I mean it's not: "OK, we are clear now!"

D. Muster:

I think that's an unfair burden to place on them. If you want them to go lower, you are going to have to justify. If you do, they'll probably do it.

F. Hart:

I think there's a great deal of variability amongst industries in terms of policies the corporate structure normally sets. There are companies in North Carolina that have set a goal of 85 dBA. They have set aggressive policies to meet that goal. I might say that with respect to the seminar programs that we present over the years at North Carolina State, we always recommend that that be the ultimate goal until such

time that a lower level is set. We have found that they will accept that as a final goal with an interim goal of 90 dBA.

S. Waggoner:

Will the knowledge and results of this study be shared with other companies? Can I get a copy of the report?

D. Muster:

No, sir. I was lucky to tell you what I did. It's not that the engineers involved do not want to share this information, it's the legal people who will not give the clearance. At one time or another virtually every company in the country is in violation, but they don't like to admit it.

C. Hurst:

Sometimes it's not the managements' fault that they can't comply. I've seen two arms of the government getting into the act and preventing a noise control procedure from being allowed. For example, I've been working with the poultry processing industry and they wanted to use a mylar covered foam material that could be placed in a hung ceiling. This one plant wanted to replace existing fiberglass lay-in panels with this foam, but the USDA would not allow it unless the joints were taped. The present joints were not taped. Thus USDA was upping the requirements, effectively preventing the use of this new material.

J. R. Bailey:

You took a swipe at researchers who do research that is not applied, and yet as a consultant you say you cannot publish. I think the point that should be made here is that universities are uniquely positioned to do research in the public domain, especially if support is provided for by, let's say, a public agency. This is the only way, I see, in which progress can be made in the public domain where results are published, distributed and implemented without the strings that often go along with other forms of support for research.

D. Muster:

Well, apparently I wasn't making myself clear. I am not about to decry the need for research, or the fact that people do it. What I tried to say was that all the research in the world, if it is not in

the hands of engineers in the smaller companies--people who do not have the specialized capability, does not do any good. These people may have an intellectual interest in research, but until the results

of the research is in their hands in a form they can use, the research does not get to where the money is; to where the products are going out the back door. This is what concerns me.

SLIDE-TAPE DEMONSTRATIONS OF THE EFFECTS AND CONTROL OF NOISE:
AN EFFECTIVE EDUCATIONAL RESOURCE

William S. Gatley, Professor

University of Missouri - Rolla
Department of Mechanical and Aerospace Engineering
Rolla, Missouri 65401

INTRODUCTION

Noise is probably the least well understood pollutant, both as to causes and control. However, recent reductions in air, water, and solid waste pollution serve to emphasize the effects of noise both as a health hazard and an irritant in the daily lives of millions.

Reduction and regulation of noise are in progress on a broad front. For the most part, however, students of all ages, professional groups, and the general public are only vaguely aware of the effects and control of noise. Without increased knowledge and support, programs to reduce noise levels will necessarily be limited.

A related problem is the current lack of educational materials that are interesting, informative, and pertinent. Universities with instructional programs and research capability in acoustics and noise control are ideally suited for developing such materials. In many cases, these same institutions can also distribute such materials to various user groups. One particularly attractive format is a presentation that utilizes 35 mm slides, coordinated with a tape recording. Some features and advantages of this format are:

- i. The presentations are self-contained and can easily be mailed for use anywhere. No prior knowledge of the subject matter by the user is necessary.
- ii. The combination is efficient, because the material has been carefully selected and uses both sight and hearing for instruction. (The latter is particularly valuable for demonstrations of types of sounds, noise levels, frequency analysis, and hearing loss.)
- iii. The compactness and self-contained nature of each presentation is adaptable for incorporation into already-crowded curriculums.
- iv. Development of the presentations does

not require an excessive amount of time or a great deal of skill. The laboratory equipment required is available at most universities with schools of engineering. Material costs are nominal.

- v. The only equipment needed by the user is a 35mm slide projector and a medium-priced reel type tape recorder.
- vi. The presentation can be interrupted at any time for questions, comments, or discussion.
- vii. Each listener can be provided with a script (including reproductions of slides), if more in-depth study (including assigned questions and problems) is desired.

SLIDE AND TAPE PREPARATION

Although the presentations must be technically correct and appropriate, they need not be of professional quality to accomplish their intended purposes.

Slides can be photographed from a wide variety of copy. One method for preparation of drawings utilizes transparencies for overhead projectors. The marking pens are available in several colors, and mistakes can be easily corrected with a cotton swab dampened with alcohol. In addition, sketches can be placed under the transparency to aid in preparation of the finished drawing. Placing a different color of poster paper under each transparency before it is photographed provides a variety of colored backgrounds for slides. Other copy includes actual noise sources, acoustic instrumentation, hearing protectors, materials and structures for noise control, etc. Slides for one of the presentations (Sources of Community Noise) depict noise sources and graphic level recordings of dBA versus time. These were prepared by photographing each noise source and graphic level recording separately. The two portions were then composed into a single

35mm glass slide and copied.

For most presentations, the dynamic range and frequency response of a typical stereo tape recorder are adequate, using a tape speed of 3 3/4 inches per second. Factors that contribute to overall quality of the tape recording are:

- i. Relative volumes of voice and other inputs (such as noise sources, pure tones, etc) must be compatible in order to provide a comfortable listening environment. Portions to be emphasized can be elevated about 5 dB above average voice levels; levels greater than 10 dB below average voice levels should in general be avoided because of possible masking by ambient noise levels in the listening area. Input levels to the tape recorder can be monitored by a sound level meter inserted in the input circuit. (The meter must be able to accept inputs from sources other than a microphone.)
- ii. For voice recording, a sound level meter can be used if the microphone is protected by a windscreens. Close-talking the microphone permits acceptable recordings to be made in relatively noisy surroundings.
- iii. Transient switching noises that occur on the tape from use of the "record" mode can be eliminated by cutting out the unwanted tape segment and careful splicing. In some cases it may be desirable to use a second tape recorder for input, or a spectrum shaper to simulate hearing or transmission loss. The resulting master tape can then be copied to obtain additional tapes for distribution.

SLIDE-TAPE PRESENTATIONS

Described below are three slide-tape presentations that have been developed over the past five years, in response to the educational needs of specific groups. Many other presentations (even a series) could be developed for other groups; the number and type depend only on the imagination, ingenuity, and experience of the author.

"LEARNING ABOUT SOUND"

Covers fundamentals of sound generation and propagation; terminology; basic instrumentation and measurements; the human ear and its response to sound; legislation; hearing loss and annoyance. Suitable for 5th or 6th grades during coverage of unit on sound; secondary school, university, and general public (as a non-technical, general-interest presentation). 23 minutes; 26 slides; script with slide changes indicated.

Representative excerpts from the script,

and examples of associated slides follow:

This demonstration is called "Learning About Sound". It is presented by the Department of Mechanical and Aerospace Engineering, University of Missouri-Rolla, through a grant from the Halliburton Company. I hope that what you see and hear will illustrate some important things about sound, and supplement what you may already know from your science and physics classes. Here are the topics we will be discussing during this demonstration:

S2

What is sound?
Why learn about sound?
How do we describe sound?
How is sound measured?
How do we control unwanted sound?
How does sound affect us?

What is Sound?

S3

Sound is a disturbance in the pressure of the air which is produced by some moving object (or source of noise) that is in contact with air. Motion of the noise source is transmitted from one particle of the air to another, creating a sound wave.

There are hundreds of sources of sound that we could name - a few are the human voice; a brass band; a lawnmower; a power saw; a vacuum cleaner; and a loudspeaker such as the one you are listening to now. I will now play a short segment of sound from each of these sources.... In each of each cases, some moving object transmits its motion to nearby particles of air; these particles in turn collide with neighboring particles, and cause the disturbance to propagate as a sound wave, in much the same way as ripples are made by dropping a stone into a pool of water....

S4

S5

S6

S7

S8

S9

How Do We Describe Sound?

S12

There are three main things that we must know about any given sound in order to determine its effects and, if necessary, to reduce it. These are the amplitude, or pressure, of the sound; its frequency content; and the speed with which the sound wave propagates....

S14

Under most conditions, a 3 decibel change in sound pressure level is about the smallest that is noticeable. This corresponds to the change that occurs, for example, when one power lawnmower is operating and a second one is started up.... On the other hand, a 10

S15

decibel increase in sound pressure level is very significant, and is judged by most people to be about twice as loud....

- S17 Another important property of sound waves is the speed with which they travel through air....

How is Sound Detected?

- S18 The presence of sound is most often determined by the ear or by a microphone. We shall discuss each of these now....

A microphone is a very sensitive device which responds to sound waves that come in contact with it. The microphone often has a thin diaphragm (like an ear drum), whose motion creates a small electrical signal. This signal can be increased in size by an amplifier, and then displayed on a meter which is calibrated to read in decibels. The instrument which does this is called a sound level meter....

- S19
- S22 Controlling unwanted sound, or noise, so that our surroundings are quieter and more pleasant, is important to all of us. Noise control always involves a source of noise, a path for transmitting the noise, and a receiver who hears the noise. Therefore, noise can be reduced by making the source quieter, by blocking the path, or by protecting the receiver. These methods usually involve the use of materials for absorbing or isolating noise. The choice of materials depends upon how the sound energy is distributed - noise sources that are mostly low-frequency, such as lawnmowers and motorcycles, require one type of treatment, while noises that have high frequencies, such as jet aircraft and power saws, require a different treatment. Since most noises are produced by machines, the mechanical engineers who design them also have responsibility for making them quieter.

How Does Sound Affect Us?

- S23 Sound and noise have two main effects on humans. These are annoyance, and loss of hearing....

- S24 If a person is exposed to loud sounds for a long enough time, he may develop a hearing loss. This means that he may not hear sounds like chirping birds or crickets, or that ordinary speech is not loud enough for him to understand. Usually, loss of hearing from exposure

to loud noises occurs gradually over a period of several years....

In 1970 the federal government passed a law to protect employees' hearing while they are at work. The law says that noise levels must be reduced so that most workers will not have a hearing loss, even after many years of exposure.

- S25 While the noise levels are being lowered, workers must wear hearing protectors which lower the sounds that reach the ear drum. These protectors can be either ear plugs or ear muffs.

Loss of hearing usually takes place gradually and without pain. In fact, the person may not even be aware that he or she is partially deaf until it is too late. In this respect, hearing loss is different from other types of injuries, such as eye damage, a cut, or a broken bone.

I will now read part of the Declaration of Independence. This is the way it would sound to a person with normal hearing... Now listen to the same passage through the ears of a person with some hearing loss. As you listen, think of how much that person is missing, and how he or she must strain to hear everyday speech. Think of how fortunate you are to have good hearing. Don't take your hearing for granted....

- S26 Here are a few simple rules that will help you to protect your hearing, and that of others:....

"AN INTRODUCTION TO INDUSTRIAL NOISE AND ITS CONTROL"

Covers sound generation and propagation; terminology; instrumentation and measurements; combination of sounds; sound fields; evaluation of data; acoustical materials; criteria and legislation; OSHA noise regulations; hearing conservation; principles of noise control; case study. Suitable for University students (particularly those in science and engineering); professional societies; technical and managerial personnel in industry. 35 minutes; 32 slides; script with slide changes indicated.

Representative excerpts from the script, and examples of associated slides follow:

- S1 This presentation was prepared at the University of Missouri-Rolla, through a grant from the Missouri Industrial Commission.

S2 The purposes of this presentation are to describe industrial noise and its effects; to discuss basic noise measurements, criteria, and legislation; and to present guidelines for, and examples of, the reduction of industrial noise.

S3 First of all, what is noise? Noise is defined as unwanted sound, and can cause annoyance, interference with speech communication, inability to hear warning signals, and even loss of hearing....

S6 Noise is characterized by its level, duration, and frequency content. The level or intensity of noise is measured with a decibel scale that typically ranges from zero (the threshold of hearing for young adults) to 120 (loud rock band, or near a pneumatic drill). The decibel scale is logarithmic, which approximates the response of the human ear to noises of different intensities....

The second reason for identifying the predominant frequencies in a noise source is that all acoustical materials used for noise control perform better at some frequencies than others. Therefore, frequency analysis is important for optimum selection of materials for a particular application.

S8 Sound level measurement and frequency analysis are obtained from sound level meters and frequency analyzers....

S9 The most common type of frequency analyzer separates the audible frequency range (20-20,000 Hz) into 10 adjoining, or contiguous, frequency bands. The lowest band is centered at 31.5 Hz, and the other bands are centered at 63, 125, 250, 500, 1000, 2000, 4000, 8000, and 16,000 Hz. Since each center frequency is twice the preceding center frequency, the resulting set of 10 decibel levels is called an octave band frequency analysis. Note that there is one decibel level obtained from each frequency band. This level indicates how much of the total acoustic energy is concentrated in a given frequency band. The resulting energy distribution or frequency spectrum is essential for evaluating and reducing noise levels, as was mentioned earlier.

To demonstrate an octave band frequency analysis, the familiar sound of a barking dog will be used....

Damage risk to hearing is assessed by A-weighted sound levels measured at the ear....

S12 However, if levels to which an employee is exposed vary throughout the working day, then the exposure can be calculated as shown by the example on the slide. Often a dosimeter or noise exposure monitor offers the best method for measuring daily exposure levels. The dosimeter is a compact sound level meter carried by the employee; a microphone is typically clipped to the collar close to the ear receiving the greater exposure. Acoustic energy arriving at the microphone is first A-weighted, and then stored in a cell for later readout. Some dosimeters have a self-contained readout, while others require a separate device for this purpose. There is also a signal that indicates when the level corresponding to 15 minutes or less exposure (115 dBA) has been reached or exceeded...

The slide shows OSHA noise limits and daily exposures....

S15 Hearing loss is determined from the average threshold of hearing at 500, 1000, and 2000 Hz, as measured by an audiometer. Regular hearing tests of employees exposed to noise levels above 85 dBA are required in order to identify those in the unprotected 15 to 20%....

S20 If an employee is exposed primarily to direct sound (such as from a nearby machine), then the addition of acoustical absorption to the surroundings will do little if any good. Remember that acoustical absorption is useful only if the receiver is exposed primarily to reflected sound, and if the material adequately absorbs the dominant frequencies....

S17 Programs for reduction of industrial noise are based upon knowledge of the generation and propagation of noise, acoustical measurements, availability of suitable materials, and above all common sense. Let us examine the principles that underlie any rational noise control program.

S18 The first step in a noise control program is to identify the noise sources present. This means not only what they are, but their importance relative to one another. It makes no sense to invest time and money in reducing noise from a source that is not significant, or

in reducing the noise to a level far below that of the remaining sources..

- S19 The next step is to list and evaluate various possible noise control procedures. In every problem, there is a source or sources of noise, one or more paths by which the noise is transmitted, and one or more receivers whose noise exposure is to be reduced. Any of these elements can be altered to correct the noise problem....
- S23 Another important noise control principle recognizes the difference between absorption and attenuation of noise....
- S24 Identification of flanking paths is another principle that is crucial for effective noise control. A flanking path is any route by which acoustic energy travels from one side of a barrier to the other, excluding the barrier material itself. Typical flanking paths include gaps around doors, windows and viewing panels, and ductwork used for ventilation, heating, or cooling. Remember that any air leaks in a structure are also sound leaks. The better the structure, the more important these flanking paths become.
- S26 A case study will now be presented to illustrate the noise control principles just described....
- S27 A noise survey with a sound level meter indicated that the two principle noise sources were the wire cutoff operation in the coiler, and air discharge from pneumatic control valves in the assembler. At a typical operator's location, coiler noise was 93-5 dBA, peak assembler noise was 100-101 dBA, and reflected sound levels from other coilers and assemblers were 92-4 dBA.
- S28 Possible noise control procedures included modification of coiler operation; enclosure of each coiler; addition of absorption to walls and ceiling; and installation of air discharge silencers on the assembler....
- S29 A frequency analysis of coiler noise showed that the acoustic energy was uniformly distributed over a wide frequency band, ranging from 200 to 5000 Hz. Because of the low-frequency attenuation characteristic of the ear, reduction of noise above 500 Hz was of greatest importance for reducing the dBA level. A reduction of 10 dBA for each enclosure was established. The total effect,

with a 10 dBA decrease in air discharge noise, would be a reduction of 10 dBA in noise levels in the coiler area, to about 85 dBA....

- S32 Of course, not all noise control problems have simple or evident solutions. However, the approach described and illustrated in this presentation offers the most effective and economical method for dealing with any noise problem, from simple to complex.

If this presentation has given you a better understanding of the nature and control of industrial noise, then its purpose has been accomplished....

"SOURCES OF COMMUNITY NOISE"

Illustrates actual and relative sound levels of typical noise sources under representative conditions. Decibel and dBA quantities are described. Each slide shows a noise source and a graphic level recording of dBA versus time. Suitable for all groups from high school through adult. 15 minutes; 18 slides; script with slide changes indicated.

Representative excerpts from the script and examples of associated slides follow:

This tape contains a sequence of 17 representative sources of community noise. The following instructions will assist you in setting up the tape recorder, loudspeakers, and slide projector so that the demonstration will be as effective as possible for your audience....

The remainder of the tape should be heard by the audience....

The sounds you will hear are neither the quietest nor the loudest that most people encounter in daily living - rather, they are representative of typical noise levels in urban or suburban areas....

The sound you hear will also be displayed graphically on a slide that illustrates the sound source. This display is on the upper portion of the slide, as you see here, and is a plot of sound level, measured in A-weighted decibels, versus time in seconds. The A-weighted decibel scale, or dBA, approximates the way your ears perceive sound. Low frequencies are weighed less heavily than high frequencies, to which the human ear is more sensitive.

CONCLUSION

Current, informative presentations about noise, its effects and control should be a part of all curriculums from elementary school through university level. In addition, suitable materials should be available to industrial, managerial, and civic groups. Universities with programs in noise control have not only the capability but also an obligation to provide suitable educational materials as a part of their educational and public service missions. Frequently, distribution can be handled by an extension division or an experiment station. Extension agents throughout the state can also publicize the availability of presentations to groups within their own areas.



Figure 3. Slide S9 from "An Introduction to Industrial Noise and Its Control".

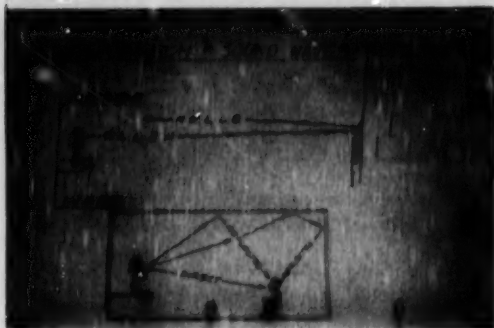


Figure 1. Slide S17 from "Learning About Sound".



Figure 4. Slide S20 from "An Introduction to Industrial Noise and Its Control".



Figure 2. Slide S19 from "Learning About Sound".



Figure 5. Typical slide from "Sources of Community Noise".

"Slide Tape Demonstrations of the Effects and Control of Noise:
An Effective Educational Resource"
presented by
W. S. Gatley

DISCUSSION:

P. Baade:

You have mentioned the importance of educating the public, at an early age, about noise. I would like to add that we should attempt to give engineers an awareness of noise control during their formal education, not to make them specialists, but simply to show them how to avoid getting into trouble with noise when designing machinery. This is not currently being done; perhaps this is why we have so many noisy machines around.

H. Nozick:

What kind of distribution is your program receiving?

W. Gatley:

To date, the program has been made available to the sponsors, and I personally have used the program. Presumably, the program could receive wider usage through the federal or local governments and through university extension services.

D. Muster:

We have within the Institute of Noise Control Engineering a committee set up to provide accreditation for noise control engineering curricula. You can contact me or Peter Baade if you are interested in this.

VEHICLE NOISE SOURCE LEVEL MEASURING SYSTEM

Elmer L. Hixson, Professor

Electroacoustics Research Laboratory
Department of Electrical Engineering
The University of Texas at Austin

INTRODUCTION

A new method for measuring acoustic noise from moving vehicles that shows considerable promise has been conceived and tested (1).^{*} Since the measured noise can be independent of distance and referenced to any convenient distance the log of the measured noise has been named the Noise Source Level. A surplus police radar, standard lab equipment, several eager students and no funds were used to implement the method and run preliminary tests.

Initial results were encouraging and the U.S. Department of Transportation funded an improved implementation and evaluation of the accuracy of the system. Comparisons were to be made to the passby measurement at 50 feet and the Society of Automotive Engineers' noise emission tests.

The sponsored study revealed that simple modifications to cheap commercially available doppler or FM radars could not provide adequate range information required for the system. In addition, sound propagation studies revealed serious noise measurement problems common to all methods for measuring vehicle noise.

NOISE SOURCE LEVEL

Definition

This new method for measuring vehicle noise is based on the long used method for specifying the source strength of an electroacoustic transducer. For example, the acoustic pressure per ampere may be measured at a convenient distance in the far field. That pressure is then corrected for distance to give an equivalent pressure per ampere at one meter. Such a measurement is based on spherical

spreading of the sound wave and a knowledge of the measurement distance. If P_0 is the equivalent pressure at one meter then the pressure at any distance is given by

$$p = P_0 / r \quad (1)$$

where r is the distance from the source in meters.

In applying this concept to the measurement of noise from a moving vehicle, its equivalent source strength is given by

$$P_0 = pr \quad (2)$$

when p and r are measured dynamically as the vehicle passes by. This can be done with a sound level meter and distance measuring radar. The spherical spreading assumption will be discussed at length later. Because of the difference in speeds for sound and radio waves a measured distance error results. If the radar distance is r' , the corrected value of P_0 is

$$P_0 = pr' / (1 - v/c) \quad (3)$$

where v is the vehicle speed toward the radar and c is acoustic wave speed.

Since decibel notation is appropriate for acoustic measurements $20 \log_{10} P_0$ has been defined as the Noise Source Level, NSL. This gives

$$NSL = 20 \log_{10} p + 20 \log_{10} r' - 20 \log_{10} (1 - v/c) \quad (4)$$

The first term is simply Sound Pressure Level, SPL as given by sound level meters. An approximation for the last term gives

$$NSL = SPL + 20 \log_{10} r' + 0.4343 v/c \quad (5)$$

For a vehicle speed of 60 mph the approximation results in an error of 0.36 dB. With a sound level meter with electrical output proportional to sound

^{*} References appear in numerical order at the end of the paper.

pressure and a radar with a range voltage output, the expression above can be implemented with simple analog techniques as shown in Figure 1.

Advantages

Such a measurement system provides some unique information and measurements for moving vehicles. As long as the vehicle noise is above the ambient, the measurement becomes a quasi-static one instead of a high speed transient. When NSL is plotted versus vehicle track angle as it is tracked by the radar, a source directivity plot is produced. With two passbys and the microphone close to the track, nearly 360° of the directivity plot can be produced. Any desired filtering of the acoustic signal can be used. The resulting NSL is a unique property of the vehicle made under the particular speed condition as if it were stationary. NSL is the equivalent sound pressure level at a unit distance that can be extrapolated into noise effected areas. The quantity provides a more accurate source model than that used in any highway or airport noise prediction technique.

Uses

Three basic uses are suggested for Noise Source Level measurements. Since it is a basic property of the source, an obvious use is for vehicle noise emission standards. It is applicable to air, water or land vehicles. This includes aircraft of all kinds, boats, highway and recreational vehicles. Measurements on new vehicles and periodic tests on old ones can be quickly made.

Noise ordinance enforcement is a second important application. Since any reference distance can be used in NSL, the one written in the local law can be used. Then microphone placement is non critical and violation levels indicated even though the vehicle is not near the microphone. Directivity effects are also included in violation levels.

Vehicle noise reduction research is a third suggested use. With frequency discrimination, accurate source levels and directivity vehicle noise sources and their characteristics can be isolated and studied. Then correction measures may be suggested and the effect of those measures accurately determined.

The suggestions above have been of a general nature. These are, no doubt, specific measurement problems to which NSL methods can be fruitfully applied.

IMPLEMENTATION

First Model, Doppler Radar

The initial system used an early model police doppler radar modified slightly to give a dc potential out proportional to vehicle speed. An ANSI type II sound level meter with an ac potential out proportional to instantaneous sound pressure gave the pressure signal. Laboratory type log converters were used and integration and summation was done with standard operational amplifiers.

The radar antenna system was mounted on a tripod so that the vehicle could be tracked in angle. This was done by hand but an automatic angle tracking method has been investigated (2). The antenna was coupled to a sine-cosine potentiometer to convert NSL to rectangular coordinates for polar plotting on an X-Y plotter. The omnidirectional microphone was placed near the radar. Pressure tubes for high-vay vehicle counting were used to start and stop the range integrator. The sign of the velocity potential was reversed at the closest point of approach with a cam on the antenna rotating shaft. Since high speed pass by measurements were not anticipated the third term in Eq. (5), the correction for vehicle and sound speed, was left off.

This system had some obvious disadvantages but one important advantage. It worked, and fairly well. Disadvantages for convenient field use included a bulky system, ac and battery power required and pressure tube installation at the measurement site. Results that will be shown later indicated that a directional microphone tracking with the radar would have been helpful. It was also discovered at this point that an automobile was a poor radar target in the pass by situation. A head on situation, typical of police speed control use, gave good return signal but close broadside aspect gave serious signal drop outs.

Improved Model, FM Radar

The improved model of the Noise Source Level Meter supported by DOT was designed to eliminate some of the disadvantages of the earlier model (3). A distance measuring radar was purchased, integrated circuit techniques and all battery operation was planned. The radar used a Gunn diode rf signal source which was frequency modulated with a triangular wave. The difference frequency between the transmitted signal and the reflected return was then proportional to range. Target motion caused a doppler shift which was removed by averaging. The system implemented with integrated circuit components and powered by a single 12 volt battery is shown in Figure 1.

The system was compact, simple to power from a car battery, simple to set up and required no pressure tubes. However, radar problems again limited the system's usefulness. In the doppler radar, signal drop outs gave zero velocity voltage and caused

some range error out of the integrator. With the FM radar, it tended to "see" the largest target, moving or not. Thus, signal drop outs from the vehicle allowed the radar to see buildings several hundreds of meters away. Then extremely large range errors resulted.

RESULTS

To test the NSL measuring system a known sound source was mounted on a pick up truck and measurements were made in a residential neighborhood (4). A horn loud speaker with a diffuser cone pointed vertically served as an omnidirectional source. Pink noise filtered to one octave centered at 1 KHz was used. The same loud speaker without the diffuser pointed horizontally served as a directional source.

Figure 2 shows the results of two passby runs in opposite directions at about 25 mph with the omnidirectional source. The "X's" on the plot are data corrected to one meter measured in an open field with the vehicle stationary. The correspondence between the two measurements is considered to be very good. Figure 3 shows the results of identical tests with the directional sound source. NSL accuracy is seen to be good to about 30° on either side of the main lobe. It is apparent that reflections from nearby houses were arriving at $\pm 50^\circ$ with respect to the main lobe. Several passing cars were measured on several occasions. Figure 4 is the complete data on a car as the result of a cooperative passer by. He made passby runs in both directions under constant speed and direction conditions. Note that the left muffler was more defective than the right one. This fact was verified by the owner. Again, reflections could be possible at $\pm 50^\circ$.

The measurement site was far from ideal yet the results were quite good. Results indicate that a directional microphone would have improved the directivity measurements at that site.

NOISE MEASUREMENT PROBLEMS

Sound propagation measurements were made in an open field to provide controlled conditions and a simulation of the vehicle noise measurement problem (5). A 1 KHz pure tone and 1/3 and 1 octave noise signals centered at 1 KHz were used. Single and double omnidirectional sources in phase, out of phase and uncorrelated were placed at a height of 0.65 meters. A measuring microphone at heights of zero and 1.5 meters over distances from 1.5 to 33 meters was used to make measurements in one 90° quadrant. Symmetry required only one quadrant. Vehicles have many noise sources but two were considered a first order approximation that provided a small enough number of variables that could be rea-

sonably controlled.

A great deal of data were taken that would take another lengthy paper to display and discuss. Therefore, only a brief verbal discussion of the results will be presented here. The first problem results from the well-known phenomena that results from a sound source above a reflecting plane being measured by a microphone also above the plane. The microphone receives a direct and reflected wave that interferes to cause deep nulls when the path difference is an odd multiple of a half wavelength. For a perfectly reflecting plan constructive interference produces levels 6 dB too high and destructive interference produces zeros. For very long ranges the two waves are in phase, the level is 6dB too high but spherical spreading controls the variation with distance.

For noise sources, as the bandwidth increases, the nulls become less pronounced because the direct and reflected wave become less correlated. When the microphone is on the reflecting plane it is at the point of reflection, pressure doubling occurs, no nulls exist and spherical spreading results. A constant error of 6 dB exists.

The measurements over imperfectly reflecting ground resulted in several maxima and minima over the distance of 1.5 to 33 meters. The 1 KHz octave band noise showed almost no max-min pattern but spherical spreading did not occur. When the microphone on the ground, no max-min pattern occurred even for the pure tone but the decay with distance in all cases was

$$p = P_0/r^{1.5}$$

instead of that of equation (1). The second term in equation (5) then becomes $+30 \log_{10} r'$. Similar results have been reported in the literature (6).

Different terrain produce different reflection coefficients which so far have defied analytical modeling. Using a ground level microphone solves one problem but different exponents are expected at different locations. Without a good propagation model these exponents are not predictable but they are measurable.

Multiple sources produce predictable directive effects but range dependence is identical to that of single sources. Thus, multiple sources present no measurement problem unless they are widely spaced so that the source-microphone distances are appreciable.

CONCLUSIONS AND RECOMMENDATIONS

It is concluded that Noise Source Level Measure-

ments can be very useful in vehicle noise problems but some problems remain. The unexpected poor radar performance is a major problem but solvable. About a two man year effort is recommended to allow many possible improvement schemes to be tried and evaluated.

The problem of nonspherical spreading and max-min patterns is common to all noise measurements and particularly important to the NSL system. However, a simple gain adjustment of the range log converter of Figure 1 can change the exponent on that term. Different sites with different reflecting surfaces are expected to give different exponents on r . The system could be used with a ground level microphone and a calibration to adjust the range term. This could be accomplished with a single pass with an omnidirectional sound source.

Because of the great potential of the NSL system it is recommended that development of the system be continued after the radar problems are solved. Again, at least a two man year effort is recommended.

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- (6) J.E. Piercy, T.F.W. Embleton and N. Olsen, "Impedance of Soft Ground and its Effect on Practical Measurements," J. Acous. Soc. Am. Vol 54, p 341, 1973.

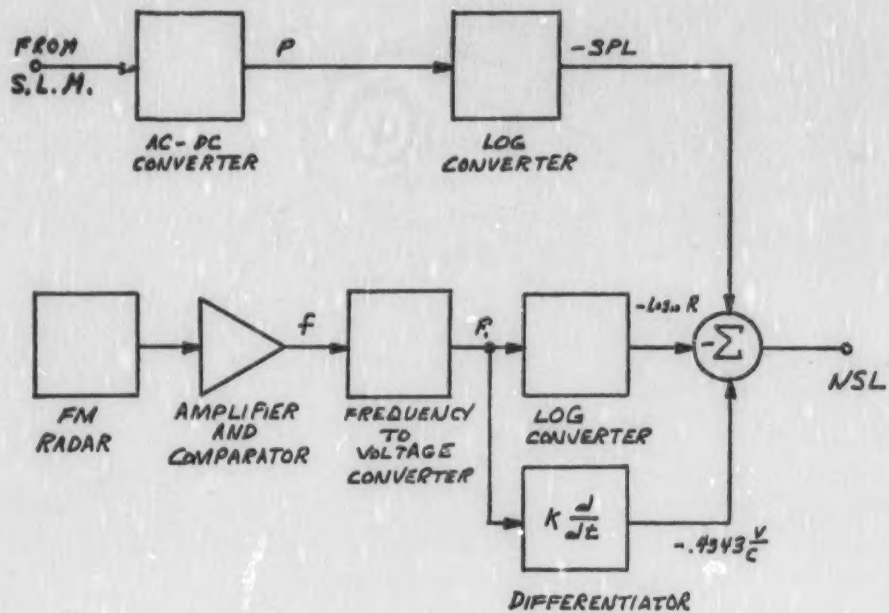


Figure 1. System Block Diagram

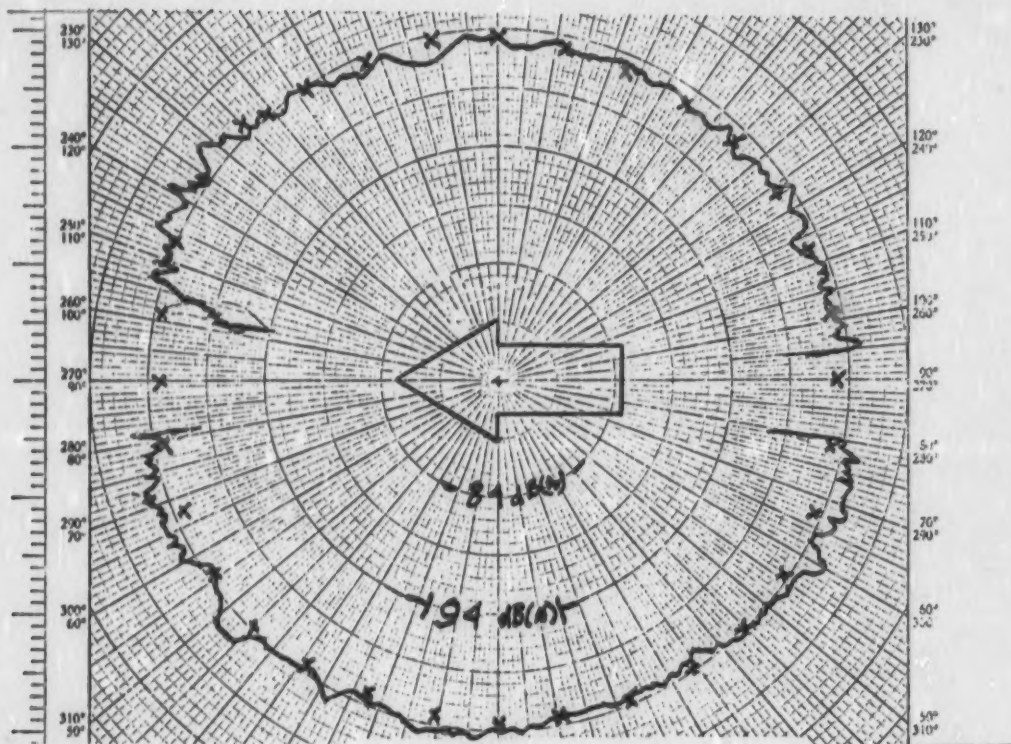


Figure 2. NSL Omnidirectional Source

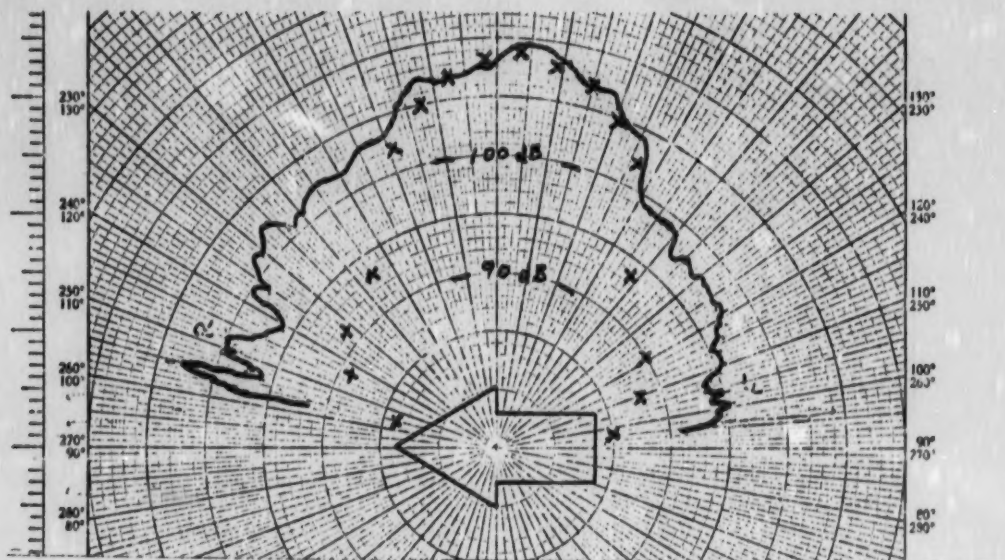


Figure 3. NSL, Directive Source

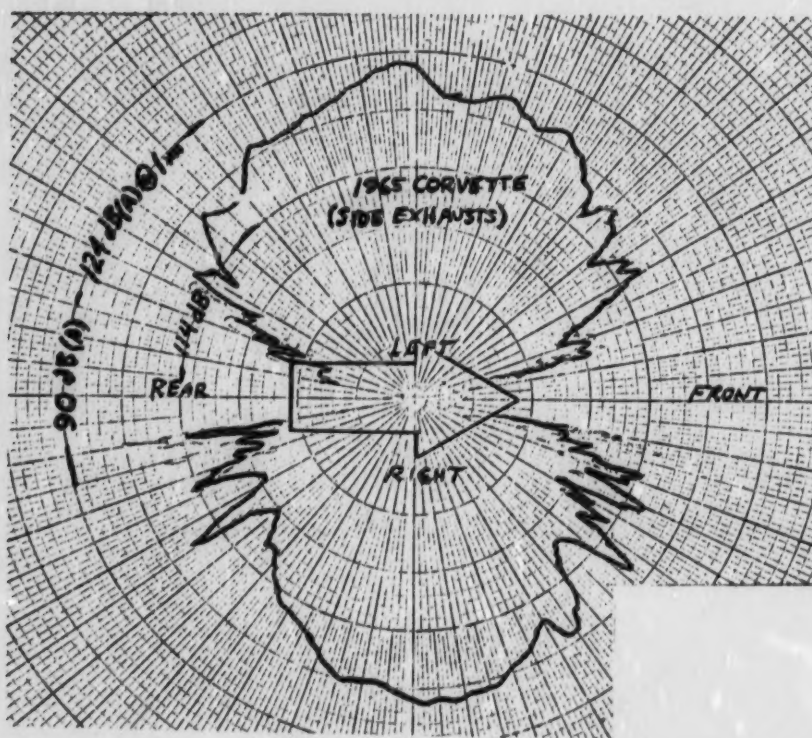


Figure 4. NSL, Moving Car

"Vehicle Noise Source Level Measuring System"
presented by
E. L. Hixson

DISCUSSION:

E. Hixson:

This project had funding of about forty thousand dollars with a total effort of 2.5 man years.

J. Sullivan:

What was the reflecting surface used to make the sound decay tests? Was it grass?

E. Hixson:

The decay data was obtained over a surface composed of a road, a patch of grass, concrete and more grass, much like the area in front of a house.

J. Gibson:

For Figure 2, what was the measurement distance?

E. Hixson:

Fifty feet.

P. Baede:

Has the ground-plane reflection problem been considered in promulgating noise legislation?

W. Roper (EPA):

This problem was approached by examining a large data base to determine the potential variability of a wide range of ground cover materials. The SAE J366 test, on the other hand, is a controlled surface test.

EVALUATION OF NEW METHODS FOR MEASURING NOISE OF HEAVY TRUCKS AND BUSES

Joseph W. Sullivan, Assistant Professor

Ray W. Herrick Laboratories
School of Mechanical Engineering
Purdue University
West Lafayette, Indiana 47907

INTRODUCTION

With the recent issuance by the Federal Government of noise rules for new trucks, the manufacturer has been placed in the role of the major enforcer. In this role, the manufacturer would do the bulk of the testing to determine compliance with the new regulations and would then label his vehicles in testimony of compliance. The role of the Federal Environmental Protection Agency in this, would be one of selective enforcement auditing.

So far, the regulations as published in the Federal Register April 13, 1976 apply only to medium and heavy trucks (over 10,000 pounds gross vehicle rating) - trucks which in many instances are custom assembled to the buyer's specification. Since the tailoring often involves major noise producing drive train components, there does not exist a "standard" production truck from the viewpoint of noise emissions. What this means is that the manufacturer will have to test frequently for compliance - perhaps 100%. This, in addition to heavy developmental loads to meet compliance in the future, implies that the truck manufacturer must have a method for noise testing that can be used on demand. Unfortunately, no such method exists.

The present method for testing is based on SAE J366b, Exterior Sound Level of Heavy Trucks and Buses. This method has gained a solid reputation with industry and regulation agencies as a means for measuring the maximum noise in dBA emitted by a vehicle, to either side, at a low enough speed where tire noise is negligible (1). The J366b test requires that the vehicle be accelerated in such a manner that maximum rated speed be attained within a prescribed distance without exceeding 35 mph (56.3 km/hr). See Figure 1 for the outdoor test pad dimensions.

Even though J366b results are well accepted, the method does have serious environmental limitations. Tests can only be conducted

under favorable weather conditions. The two weather conditions which most adversely affect tests are precipitation and wind. The J366b standard recommends specifically that the test pad be dry during the tests and that the wind speed be less than 12 mph (19.3 km/hr). A wet pad surface increases tire noise and rain increases background level noise in addition to having unpredictable effects on dissipation and scattering of sound wave propagation. Also, the presence of both rain and wind require special microphone protection which introduce special problems in calibration and maintenance.

The effects of wind on sound propagation between source and receiver are complex and unpredictable in practice. The effects cannot be gaged solely in terms of an average wind velocity but must take into account the detailed nature of the velocity distribution close to the ground, that is, in the boundary layer (2). The general effect is one of diffracting, or bending of the sound waves in proportion to the local velocity gradient, which in turn alters both the phase relationship and amplitude of the sound waves at the microphone.

Furthermore, variations in temperature, humidity, and barometric pressure can produce unpredictable effects on sound propagation from the vehicle to the microphone, and changes in the microphone itself which cannot always be corrected. The variable and unpredictable nature of the weather in the Midwest (where most heavy trucks are produced) limits both the scheduling and the number of tests which can be conducted. On the average only 75% of the working days can be used for testing. And much of this must be done using microphone wind screens. It becomes mandatory, therefore, to have a method for test independent of weather conditions.

In recognition of this need for an all-weather test method, the Motor Vehicle Manufacturers Association (MVMA) has initiated

research at the Herrick Laboratories, Purdue University, to determine what methods are feasible for all-weather testing.* This paper describes two such methods.

TEST METHOD REQUIREMENTS

Aside from the need for weather protection, a test method should have the following major requirements:

1. The noise data must be compatible with the end use.
2. The measurement facility should be located close to the manufacturing area.
3. The time for test should be short and the manpower required should be minimal.

In essence, an all-weather facility means a fully enclosed area capable of some temperature and humidity control. Clearly, it is not practical to enclose a J366 site even if the vehicle were located stationary and the microphones moved past the vehicle, as in the International Harvester SNAP facility (3). The SNAP (Source Noise Analysis Pad) facility makes use of the concept of maintaining the same relative displacement between microphone and vehicle as a function of engine speed as in the SAE J366b test, but with the vehicle stationary. This saves considerable space since the running and shutdown distances for the vehicle are eliminated. Even so the space required is on the order of 11,000 ft² (1020 m²). The cost to fully enclose, acoustically treat, and maintain such a facility would be prohibitive. Then too, a large facility could not, in general, be fitted in next to existing manufacturing areas.

It is seen therefore that an all-weather test method to be considered practical must encompass a relatively small space. This means that the vehicle must be located stationary and the microphone(s) must be placed much closer to the vehicle. This leads us directly to the question of what kind of measurement data such a scheme would produce. In the SNAP method, it has been shown that the max dBA value correlates well with J366b data. This could be anticipated since both measurements are made at the same point in space relative to the vehicle, at a given engine speed, even though the acceleration-time history of the vehicle is not the same. However, because of the reflective ground plane, one would not expect correlation when microphone orientation is changed because of the dependence of the acoustic field on the noise source distribution and spectral characteristics of the vehicle. This dependence

* The opinions, findings, and conclusions expressed in this paper are those of the author and not necessarily those of the Motor Vehicle Manufacturers Association, Inc.

reveals the weakness of the J366b method: sound pressure is obtained at only one point in space, and, although undoubtedly useful for determining maximum roadside noise, there is as far as is known no technical basis for using this information to extrapolate to farther distances.

Thus, the fundamental question involving measurements in a small space is should the test method be designed to simulate a drive-by test, that is SAE J366b, and thereby correlate with an established, existing data base; or should the method be designed to provide another, possibly more useful measure of noise such as sound power? The two methods chosen for study address both viewpoints.

DESCRIPTION OF NEW METHODS

Figures 2 and 3 illustrate the basics of the two methods under evaluation. Both methods utilize a semi-anechoic room and multiple microphones in sampling the acoustic field of the vehicle as the vehicle simulates an operational mode.

In the "Reduced Drive-By Simulation" method, Figure 2, the microphone arrays (one on each side for a 2-side test) are traversed past the vehicle as the vehicle simulates a wide open throttle acceleration. This is a direct adaptation of the SNAP method except that multiple microphones are used for space averaging of sound pressure in the vertical direction and the microphones are at a distance of 15 feet (4.6 m) from the vehicle centerline versus 50 feet (15.2 m) in the SNAP method. By keeping the same angular orientation of the microphone array with respect to the vehicle as in the SNAP method (or the J366b method) during acceleration, the acoustic directionality is more or less preserved depending on whether the vehicle has extended sources, such as a front engine chassis with rear exhaust.

The effect of microphone spacing for this concept was examined in a preliminary study by a simplified multi-pole computer model of a typical COE truck tractor with vertical exhaust (4). Figures 4 and 5 show a comparison of the microphone (receiver) sampling of the field directly out from the side of the vehicle when the source characteristics are broad band (Figure 3) and narrow band (Figure 4). A study of these two cases reveal that (a) the space average far field begins at about 15 feet from the "vehicle" centerline, (b) adequate space averaging can be done with only a few receivers and (c) the degree of correlation between single receivers using inverse square law depends on the bandwidth of the source: the wider the bandwidth the better the correlation.

In the "Semi-Anechoic Sound Power" method, Figure 5, the sound power of the vehicle is obtained, at a fixed speed/load condition, based on the theory of measuring sound power in a semi-anechoic space (5). In this adaptation, microphones are arranged along a circular boom that, when rotated, would cut an imaginary hemisphere into equal areas (6). For a hemispherical surface located in the far field of the vehicle, the sound power is calculated from the equation:

$$L_w = \bar{L}_p + 20 \log r - 2.5 \text{ dB, re } 10^{-12} \text{ watt}$$

where $\bar{L}_p$ is the level (re $2 \times 10^{-5} \text{ N/m}^2$) of the space average sound pressure over all microphones, and r is the radius of the hemisphere in feet.

The measurement procedure would consist of a continuous scanning and averaging of the microphone array, in each 1/3-octave band, as the boom rotated at a constant speed. At the end of one revolution, the final average level, $\bar{L}_p$, in each band would be used to calculate the corresponding L_w from the above equation. Alternately, the value of $\bar{L}_p$ by itself, after A-weighting could be used as the measure of the vehicle noise since it would represent that L_p which would have been measured at radius r had the noise from the vehicle been radiated uniformly in all directions.

As a final note, the directivity pattern of the vehicle could be extracted from the measured data if a storage capability were available as part of the measurement system.

CURRENT PROGRAM STATUS

Neither method has been evaluated as of this writing. The major effort to date has been in the construction and checkout of the microphone traverse mechanisms; and in measuring and defining the acoustic field around a typical COE truck. Figure 7 shows the truck and the vertically suspended microphone boom to be used in the simulated drive-by measurements. The boom is synchronized to the truck in such a manner that the displacement of the boom is controlled by engine speed. Engine load is controlled by adjustment of water flow to a water brake dynamometer mounted in the truck behind the transmission.

Since both methods require multiple microphone sampling, the instrumentation has been designed to accommodate both. A schematic of the system is shown in Figure 6. The programmable calculator (minicomputer) will serve to control microphone sequencing, store and manipulate data, and provide both numeric and plot output.

The current level of effort involves two Masters degree students, and the program timetable calls for this level of effort through 1977.

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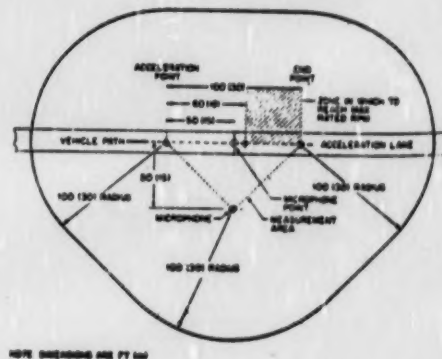
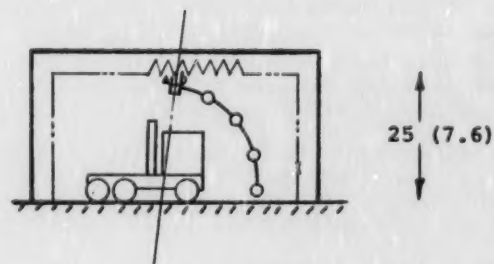
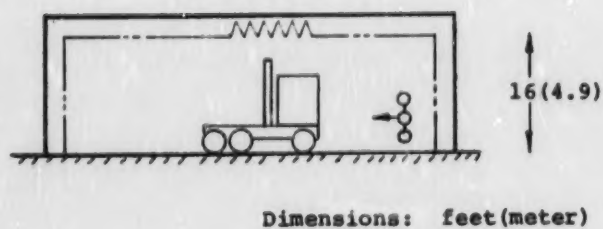
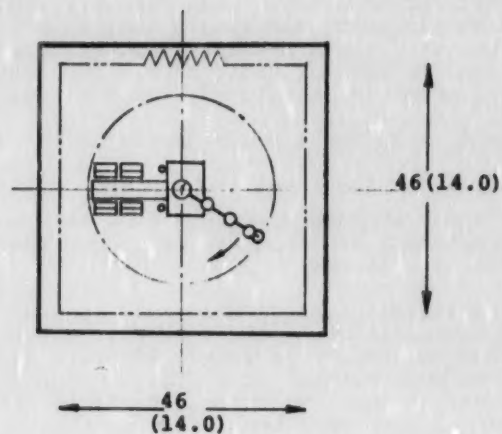
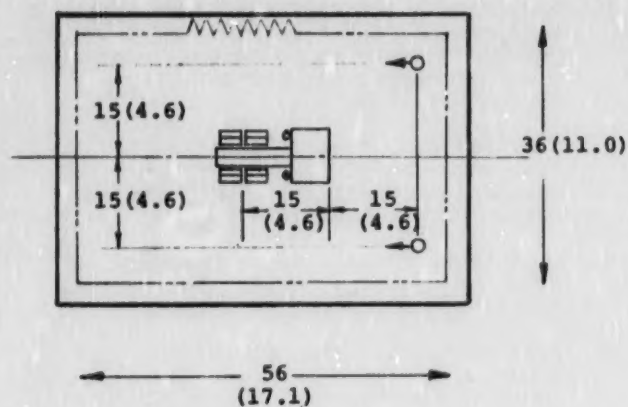


Fig. 1 Minimum SAE J366b Test Site



Dimensions: feet (meter)

Fig. 2 Reduced Drive-By Simulation Method

Fig. 3 Semi-Anechoic Sound Power Method

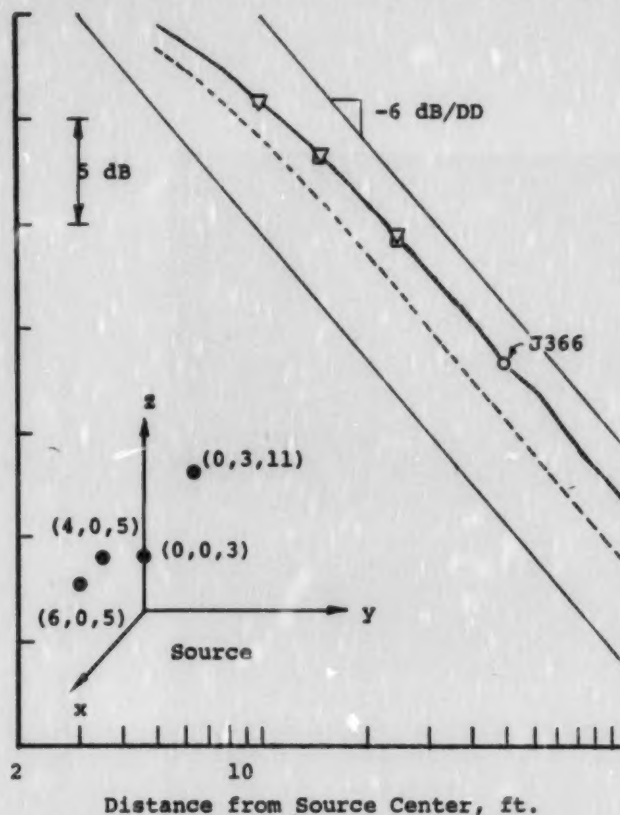


Fig. 4 Field Decay as Function of Receiver Average for Broad Bandwidth 4 Point Incoherent Source

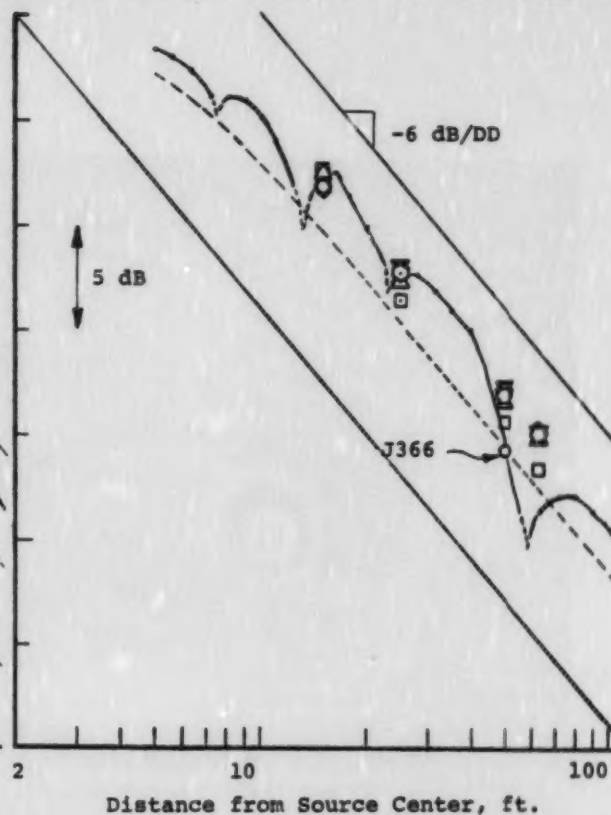


Fig. 5 Field Decay as Function of Receiver Average for Narrow Bandwidth 4 Point Incoherent Source

- Single Receiver - 4 ft. above ground
- Single Receiver - ground plane removed
- 2 Receiver Average
- △ 4
- 8
- ◇ 16
- ▽ 32

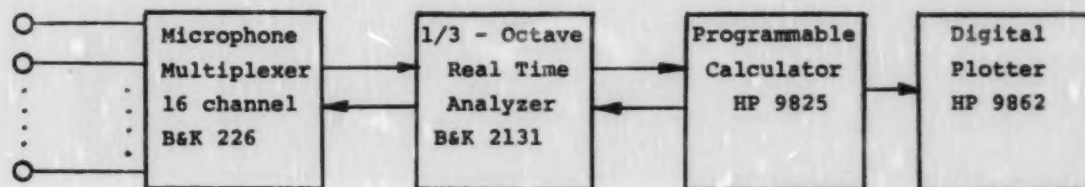


Fig. 6 Schematic of Instrumentation



Fig. 7 Vertical Microphone Boom and Truck in Semi-Anechoic Room

"Evaluation of New Methods for Measuring
Noise of Heavy Trucks and Buses"
presented by
J. W. Sullivan

DISCUSSION:

J. Sullivan:

The project I have described is funded at a rate of sixty thousand dollars with a level of effort of one and one-half man years.

R. Lyon:

This test (SAE J366) of measuring pressures around the side of a source is one of the worst ideas ever generated. For a source in a multi-path environment, one really wants to know the sound power of the source.

What is wrong with driving a truck through a reverberant room with open ends?

J. Sullivan:

It's a practical problem of climate control and expense. Also, low frequencies are a problem in reverberant rooms.

J. Tichy:

Reverberant rooms are not as expensive as semi-anechoic rooms, and if the room is large enough, low frequency measurements are not a problem.

J. Sullivan:

An additional benefit in using a semi-anechoic environment is that directivity information can be obtained.

PREDICTION AND MEASUREMENT OF HIGHWAY NOISE

Rinaldo Greene, Assistant Professor

School of Engineering
The University of Alabama in Birmingham
Birmingham, Alabama 35294

INTRODUCTION

In recent years there has been an increased public awareness of noise. Traffic noise in particular has steadily grown with the general growth of the population, and with it, concern over its intrusion. The increasing number of vehicles and their greater concentration on major highways are major contributing factors toward this problem.

In February 1973 the Federal Highway Administration of the Department of Transportation, in recognition of the problem, presented new governing regulations in its Policy and Procedure Memorandum PPM 90-2 entitled Noise Standards and Procedures. This memorandum required each state desiring federal aid to achieve noise levels on all its highway projects below certain standard values. For example, exterior to residences, schools, churches, hospitals, parks and a number of other locations the design noise level was set for an L_{10} value of 70 dBA.

This means that in these locations the value of 70 dBA could not be exceeded more than ten per cent of the time, even during rush hour.

The State of Alabama Highway Department recognized that it must develop the capacity to accurately predict traffic noise adjacent to planned highways. This would involve not only the implementation of proper analytical techniques, but also the development of experimental procedures and apparatus sufficient to measure in the field the needed statistical description of traffic noise.

The overall purposes of the contract between the University of Alabama in Birmingham and the State of Alabama Highway Department were the development of analytical and experimental tools to permit the accurate prediction and measurement of highway traffic noise. The actual goals which were achieved were:

- the identification and implementation of an automatic traffic noise data acquisition and analysis system. The system is of ANSI Type 1 accuracy and capable of determining the statistical noise parameters currently required by federal regulations.
- the development of a detailed user's manual for the automatic traffic noise data acquisition and analysis system.

tion and analysis system.

- the verification of the adequacy of the equipment system and the experimental procedures by carrying out field work at a number of sites in conjunction with personnel from the Alabama Highway Department.
- the development of a fully verified and documented computer program for the prediction of traffic noise descriptors based on the methodology of the National Cooperative Highway Research Program (NCHRP) Report 117 and including the corrections presented in NCHRP Report 144.
- the preparation of a detailed user's manual for the traffic noise predictive program, including guidelines for the selection of various, somewhat subjective, input parameters.
- the implementation of the predictive program of the Transportation System Center and an evaluation of the adequacy of its documentation from the viewpoint of a new user.
- the comparison of measured and predicted traffic noise parameters and a statistical analysis of the observed differences.
- the identification, collection, and cataloging of those published materials which reflect the present state of the art for traffic noise prediction and measurement for the Alabama Highway Department.

The final three volume report HPR Report Number 78-A is available from the National Technical Information Service in Springfield, Virginia.

PREDICTIVE PROGRAMS

There are two noise prediction methods that are approved by the Federal Highway Administration. The method contained in National Cooperative Highway Research Program Report 117 is the one on which the emphasis was placed during this study. The other approved method is contained in the Department of Transportation, Transportation Systems Center Report DOT-TSC-FHWA-72-1. All other noise predictive methods or variations of the two above require special approval.

First, consider the Transportation Systems Center (TSC) method. The computer program is available in the form of a deck of punched cards with documenta-

tion from TSC in Cambridge, Massachusetts. At the University this program was adapted for use by the Highway Department's computer and several test cases were run. The documentation was found to be adequate for a new user and the results were comparable to those from the NCHRP 117 method.

The greatest part of the effort at the University centered on the NCHRP 117 method. The algorithm for this predictive scheme was developed by Bolt Beranek and Newman in 1971 as NCHRP Report 117 entitled Highway Noise, A Design Guide for Highway Engineers. It was followed up in 1973 by NCHRP Report 144 by the same consultants. This later report presented modified procedures of evaluating the noise reduction qualities of roadside barriers, elevated and depressed highway sections, roadside structures, and such factors.

The Alabama Highway Department had in its possession a computerized version of the NCHRP Report 117 method and had applied it on a number of occasions to predict traffic noise. The results were compared to actual measurements which were made using a simple manual check-sheet procedure. The values were found to be often as much as 8 dBA apart. The highway engineers consequently had little confidence in the technique. The variability, of course, might have originated from several causes. Part may have been due to the imperfections in the Report 117 method as the program in use did not have the Report 144 corrections. No documentation was available for the computer program so that it was possible that part of the error stemmed from either errors in the program or from the misuse of the program itself. Finally there was reason to believe that part of the variability was due to the measurement method.

NCHRP Report 117 Computer Program

It was an express purpose of the program at the University to fully document the Highway Department's version of the NCHRP Report 117 computer program to ensure that it was free of error. Of equal importance was the goal of making the program easy to use even by inexperienced personnel.

The coding of each step in the program was carefully checked. The approximate representation of the graphs in Report 117 used in the computer program were verified and, in most cases, extended in the interest of greater accuracy. All symbols used in the entire program and their units if they were dimensional were fully identified by the extensive use of COMMENT cards. Each symbol, even if used as a dummy variable, was used to represent only a single variable throughout the entire program.

The input and output formats were improved for purposes of clarity and to reduce the chance of coding errors. A pre-printed input data form was prepared. All variables used as input variables were discussed in detail. Special attention was given to those input parameters which seemed somewhat subjective in nature. As an example it was necessary to define such terms as "Normal", "Smooth", and "Rough" roadway surfaces so that the proper correction factor could be applied. In the absence of specific information on such matters different engineers were

likely to use different definitions.

In order to illustrate typical program input and output, ten highway configurations were conceived and analyzed using the computer program. Furthermore, all ten cases were completely checked and verified by hand calculations. A copy of the completed input form, a description, a sketch, and a computer printout was given for each of the ten cases.

Detailed instructions for the program's use, including the ten test examples, and a listing of the final program itself is available in Volume III, User's Manual for Traffic Noise Prediction Computer Program HPR Report Number 78C.

It should be emphasized that no changes were made to the program which would result in its not conforming to the authorized predictive methodology.

The program was used to predict the traffic noise parameters at 66 sites where field measurements were taken. The 117 program was found to be an excellent predictive program for Alabama Highways in general and was recommended as the routine analytical tool of the Highway Department.

HIGHWAY NOISE MEASUREMENT

Having an error-free, well-documented, and authorized computer program was important to the Highway Department. However having a suitable technique for measuring noise was essential as well. The Department was very cautious, and with reason, in believing the output from predictive computer programs. Without having the ability to make reliable measurements which agreed with predictions for existing highways how could they really have confidence in their projections for noise levels on highways to be built twenty years hence?

EQUIPMENT SELECTION AND TESTING

Clearly the selection of good instrumentation was a top priority. It was an early decision that the equipment be of high precision and capable of achieving ANSI Type 1 specifications. This eliminated a number of potential systems from consideration. In evaluating various traffic noise measurement and analysis systems the following guidelines were adopted:

- the equipment must be capable of measuring those noise parameters incorporated into current federal regulations. It should also be able to determine those noise parameters which might be incorporated into future federal and state regulations.
- the equipment should be capable of automatically producing data or records which can be cataloged and permanently stored.
- the equipment should be in actual production and have been subjected to widespread and pertinent field maintenance.
- the equipment should have a good maintenance record and the manufacturer should have a good reputation for service.
- the equipment should be available for demon-

stration purposes and the manufacturer should supply a list of customers who can be contacted.

The equipment search resulted in the identification of two systems manufactured by different firms which would satisfy the principle requirements. The DA 600 Digital Acoustics system was selected.

An important objective of the study was the documentation of the noise data acquisition system so that even an inexperienced person could quickly learn to operate it. The details of this documentation are given in Volume II, User's Manual for DA 600 Noise Data Acquisition System. This volume includes a lengthy discussion of all components of the system and cook-book procedures for its use.

In recognition of the fact that proper planning greatly simplifies measurements in the field attention was given to this aspect of the data acquisition procedure. Special rugged carrying cases were made to accommodate each of the two sets of data acquisition equipment. As another example of field oriented activity, a substantial amount of practical information concerning the protection of condenser microphones in high humidity environments was presented. The intent of these efforts was to help minimize the difficult problems that are always present when field measurements are being made.

In order to check the various equipment components and to develop good operating procedures 66 field tests were conducted with, and sometimes by, personnel from the Alabama Highway Department. All components and equipment configurations were determined to be fully operational and to accomplish the intended purposes.

ACCURACY OF MEASUREMENT RESULTS

Although no measurement is totally error free, the equipment system and utilization procedures developed for the Alabama Highway Department is as error-free as is practical with current technology.

The precision microphone and the electronic circuitry complies with ANSI Type 1 specifications. Hence the individual readings are generally accurate to within ± 1 dBA.

The data readings are acquired at the rapid rate of one per second so that many readings can be acquired during even a short run and L_{10} can be accurately measured.

The dBA readings are encoded in binary form thus preventing the degradation that might result from recording the data in analog fashion.

There is no loss of accuracy in running the analysis for L_{10} on the associated Wang calculator since the programs do little more than count, store, and print.

OPERATIONAL FIELD PROCEDURES

As some error will always occur in using the DA 600

system and in making related traffic measurements, good field technique must be strictly adhered to.

The sound level meter must be calibrated before a run. It must be protected against hostile environments such as rain, excessive humidity, and extremes of temperature. Of course it should never be dropped or treated roughly.

During a run, all extraneous noises such as conversation, barking dogs, trains, and airplanes must be rigorously excluded from the reading. The desired measurement is the L_{10} value of the traffic noise.

Should such extraneous noises occur, the recording can be stopped or the unwanted noises can be removed during the analysis using the software designed for that purpose. The subjective channel of the DA 600 is useful for locating these noises.

The use of radar to measure speeds and the use of traffic counters to determine traffic flows of car and trucks is recommended.

Careful measurements of highway alignment and important distances is important.

CONFIDENCE LIMITS FOR L_{10} MEASUREMENTS

L_{10} is a statistical descriptor. It could be estimated a number of different times for the same basic traffic flow and a number of different values would result. What one is interested in is estimating its "true" value for the given traffic flow distribution. This is much the same as trying to estimate the true mean of some distribution by calculating averages based upon a limited number of measurements. In general the more values which go into the calculation of the average the more likely the average will be close to the mean. In a similar fashion L_{10} is estimated by analyzing numerous measurements of the noise level. The more measurements that are made the more likely the estimated L_{10} value will be close to the true value.

Detailed calculations were made which showed that at a data acquisition rate of one measurement per second the DA 600 system produces estimates of L_{10} which are accurate within ± 1 dBA with 95% confidence if the run is 15 minutes long or longer. The theory and calculations and results for both L_{10} and L_{50} at 95%, 90%, and 80% confidence levels are presented in Appendix A of Volume I of the final report.

COMPARISON OF MEASURED AND PREDICTED VALUES

It has been stated earlier that noise measurements were made at 66 locations. For each of these the L_{10} value was predicted by the NCHRP Report 117 program in its final form and comparisons were made.

In 28 of the 66 cases one or more input variables were outside the range of the charts in NCHRP Report 117. For example, the traffic flow or the

percentage truck mix at times were less than the smallest values appearing on the charts. The computer program was coded to assume the smallest chart value whenever such occurred.

Since it was reasonable to expect a priori that the statistical characteristics of the data might depend on whether or not all input parameters were within range three sets of statistical analysis were performed: on just the 38 cases within the range, on just the 28 cases outside the range, and on all 66 cases lumped together.

The findings were essentially the same for all three groupings.

The average difference between the measured L_{10} value using the data acquisition system and the predicted value using the corrected NCHRP Report 117 program for the case of 66 measurements was 1.91 dBA, the predicted value being the higher.

Three tests for normality were applied to the set of 66 values of Δ , the predicted value minus the measured value. They were the chi-squared goodness-of-fit test, a skewness test, and a kurtosis test. At the 95% confidence level, the data did not contradict the hypothesis of normality for any of the three tests.

Although the mean of the 66 values of difference ($\Delta = \text{predicted} - \text{measured}$) is 1.91 dBA it was found that the probability is 95% that the actual mean falls somewhere in the interval from -1.62 dBA to +3.44 dBA. Thus, the data collected with the 66 runs do not contradict the possibility that on the average the predictions give the same values as the measurements.

Modifications to the Report 117 predictive methodology cannot be justified by the 66 tests discussed above. A larger number of tests with a wider range of traffic and highway characteristics would be required for the development of any correcting changes. And indeed, the result of a more extensive test program might well be that no changes are justified.

From the 66 tests, the best point estimate is that the predictive program overpredicts by about 1.9 dBA. However, there is a substantial probability that the difference is less than this, perhaps even zero.

Based on the 66 tests conducted in conjunction with this report, it appears that the NCHRP Report 117 methodology predicts traffic noise with an accuracy consistent with that of most engineering design theories involving complex problems.

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Research Project 930-075

"Prediction and Measurement of Highway Noise"
presented by
E. R. Greene

DISCUSSION:

P. Mansbach:

Since you sample once per second, aren't these samples correlated? Doesn't this invalidate your calculation of confidence limits?

E. R. Greene:

Yes, the data points are somewhat correlated, but I don't think this is a problem.

R. Lambert:

I'm wondering if the various highway departments around the country talk with

one another and how much money has been expended by them modifying the 117 procedure.

I don't think the accuracy that the computer programs give is really needed.

How is the highway department using this data?

E. R. Greene:

To prepare for anticipated lawsuits, and to find the source of errors in noise measurements.

ELEVATED MEASUREMENT OF URBAN TRAFFIC NOISE

S. E. Froseth and R. F. Lambert
Department of Electrical Engineering
University of Minnesota
Minneapolis, Minnesota 55455

INTRODUCTION

Several researchers (1,2) have suggested that high-altitude noise measurements might be an effective means of establishing baseline data for a spatially averaged noise exposure in urban areas. The effectiveness of noise abatement regulations in a specific area could then be assessed by repeating aerial measurements over a period of time. Measurements made at an elevated point above a noise-impacted area are not subject to the difficulties that are usually associated with ground-level measurement technique for obtaining spatial averages. Among the usual ground-level technical problems are: 1) a large number of measurements must be made, 2) measurements should be made simultaneously, and 3) the repeatability of measurements at a particular site can be strongly influenced by changes in the local noise environment.

The main objective of the research here reported was to extend our understanding of and experience with aerial noise measurement methods. It was decided to focus the study on the sound field above a special type of urban area -- a large metropolitan downtown commercial area, having "canyon-like" streets with hard asphalt, concrete, and glass surfaces. This type of urban area will hereafter be called a "reverberent city". Since motor vehicles contribute most of the ambient noise in such a situation, this study focused on relationships between the density of vehicular traffic in the downtown area and the sound pressure statistics at an elevated measurement point. Parameters of primary interest were the heights of the buildings and other geometrical dimensions of the city, the height of the measurement point, the effective area of coverage of the measurement, and the influence of atmospheric absorption. The investigation involved the formulation of a new theoretical acoustic model of a reverberent city; the models were verified on a limited basis with appropriate experimental measurements.

This paper is divided into four parts: (1) a summary of the theoretical models, (2) a comparison of experimental results and theoretical predictions, (3) a summary of the findings of the study and (4) an estimation of effort.

THEORETICAL MODELS

Two theoretical models for representing the sound field above a city were developed. Several assumptions were made about the sources of noise and the city in which they operate; these are stated in the description of each model.

The Flat City Model

The first model, called the "Flat City Model", is similar to a derivation for steady-state noise at ground level by Shaw and Olson (3). Consider the city to be a plane, circular region of finite radius R_c with N sources (motor vehicles) operating per unit area, each having acoustic power output W in a given band of frequencies. The parameter N is constant over the region and the sources act as statistically independent monopoles located on an infinite, perfectly reflecting plane surface. The directivity factor of each source is independent of the angles of azimuth and elevation; that is, the source power is radiated uniformly into a hemisphere above the plane of the city.

Figure 1 shows the cylindrical coordinate system that is used. Using this model, the total mean-square pressure at the measurement point O was found to be (4)

$$\overline{p^2} = \rho c W N \{ -E_1[-\alpha_n z_m] + E_1[-\alpha_n (R_c^2 + z_m^2)^{1/2}] \}, \quad (1)$$

where ρc is the specific acoustic impedance of air, which is taken to be approximately $\rho c = 415$ rayls for the purposes of this study. In (1), z_m is the height of the measurement point and R_c is the radius of

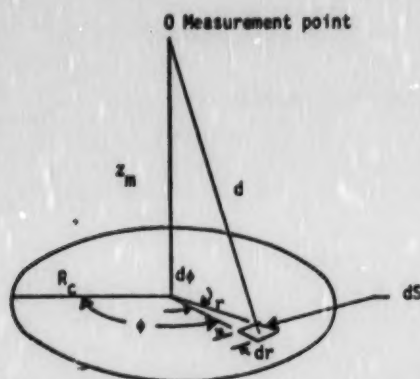


Figure 1. Coordinate system for Flat City model.

the circular city. The quantity $E_1(x)$ is the exponential integral. The parameter N may be estimated by counting traffic on city streets, or by other means. The parameter α_n is dependent on temperature, relative humidity, and the frequency range of interest, and may be obtained from one of several references. Values for the noise power W , which depend on frequency and the type of vehicle, are also available in the literature.

The Reverberent City Model

The formulation of the second noise model is in some respects similar to that of a model for "Noise Propagation in Cellular Urban and Industrial Spaces by Davies and Lyon (5). The model differs, however, from the one in Ref. (5) in that it specifically describes a hard-walled, canyon-like downtown area, and includes several simplifying assumptions. Here the downtown area is modeled as an array of semi-reverberent cells corresponding to the street and sidewalk spaces between building. There are three types of cells: "A-cells" (street intersections), "B-cells" (east-west streets) and "C-cells" (north-south streets). The city is assumed to be square, having uniformly square buildings and uniform street widths. A 3 by 3 block city is shown in Figure 2.

The height of each building is the theoretical city is assumed to be h , some mean value. The surfaces of the streets and building walls are assumed to be perfectly reflecting, that is, their absorption coefficients are zero at all frequencies. This is the so called "ideal reverberent city". The nature and location of the motor vehicle noise sources in this model are identical to those in the Flat City model, except that the sources within the city are uniformly distributed over the streets only.

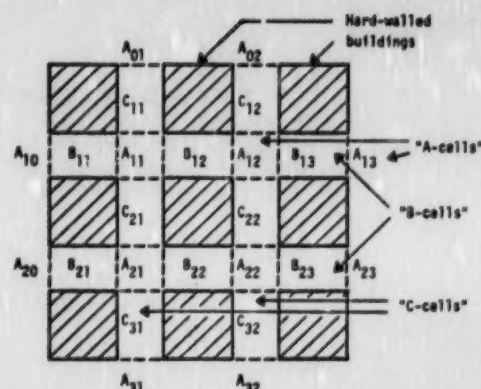


Figure 2. Urban cell model (top view).

The total mean-square pressure at the aerial measurement point may be expressed as the sum of a "direct component" and a "reverberent component". The direct component is the sound field at the measurement point that results from line-of-sight propagation from source to receiver; the reverberent component is the sound field that is produced at the measurement point by multiple reflections from the streets and building walls, that is, the portion of the reverberent field that passes through the tops of the cells. The procedure used to compute the contributions of these components is divided into three parts: (1) calculation of the reverberent sound field at street level using power-balance relations; (2) computation of the contribution of the reverberent field to the total mean-square pressure at the aerial measurement point, using a simple radiation model, and (3) using a similar radiation model, computation of the contribution of line-of-sight propagation to the total sound field at the measurement point. The results obtained in (2) and (3) are the reverberent and direct components, respectively. They are summed to obtain the total mean-square pressure of the sound field. The calculations were done in a straightforward manner using a digital computer program termed CITNOIS.

The two theoretical models yielded similar results for situations approximating the anticipated experimental measurements. However, each model in itself provided insights into various characteristics of the aerial measurement technique.

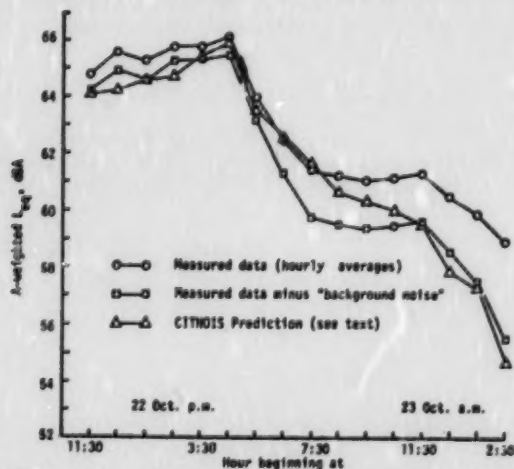
EXPERIMENTAL

To test the practicality of the aerial noise monitoring method and to verify the theoretical models on a limited basis, two experiments were conducted. Tape recordings of noise above the central business district of Minneapolis, Minnesota were made on two separate occasions from

the tops of two of the city's tallest buildings. The first set of tape recordings was made on October 22-23, 1975 from a window on the 28th floor of the Foshay Tower, about 90 m above street level. This paper reports some of the results obtained from this set of tape recordings.

A precision sound level meter with a wind-screen protecting the microphone was used to provide an audio signal to the input of a high-quality portable tape recorder. Beginning at 12:00 noon, recordings three minutes in length were made at 15 minute intervals until 3:30 a.m. on the following day. A calibration tone and voice identification were placed on the tape at hourly intervals. Noise signals on the tape were A-weighted and analyzed with a statistical distribution analyzer, enabling the computation of the energy-equivalent sound pressure level L_{eq} for each three minute noise sample (6). These measured L_{eq} values were compared with values predicted by the CITNOIS model.

Noise level predictions with the CITNOIS model required suitable geometrical parameters describing the dimensions of the measurement situation. Values for the acoustic power output of different types of motor vehicles were obtained from the literature, and the vehicle density for each type was derived from traffic flow data provided by the Minneapolis Public Works Department. The prediction also included the effects of atmospheric absorption, dependent primarily on air temperature and relative humidity. It was necessary to first perform the predictions in octave bands, and then combine the results to obtain the A-weighted level prediction.



Improved theoretical models for the reverberant field that include the frequency-dependent effects of sound absorption during propagation and reflection could be developed. Other refinements might include accounting for the random movement of vehicles and the random heights of buildings. More experimental work remains to be done in various areas of urban noise measurement before elevated surveys can be utilized. The repeatability of the method should be tested. Further verification of the theoretical models would be facilitated by large numbers of measurements from different heights and under varying traffic conditions. Correlation of noise levels and traffic densities could be done for different days of the week and for different seasons. A sufficient amount of data would make possible an accurate "calibration" of a theoretical model and also provide a data base with which future measurements and trends could be compared. The method could possibly be extended to industrial areas involving other kinds of noise sources.

Another interesting application of this measurement method is the development of relations between aerial noise measurements and actual noise levels on the ground. Experimental approaches could employ simultaneous ground and sky measurements or measurements using scale models.

EFFORT

(A) - The above traffic noise study was undertaken over a 12 month period with roughly the following breakdown in effort and equipment costs:

Task	Level of effort/ man months		
(1) Theoretical Modeling	1 ⁺	4 ⁺⁺	0 ⁺⁺⁺
(2) Field Measurements	0	1	0 ⁺⁺⁺
(3) Computer work	0	1	0 ⁺⁺⁺
(4) Equipment	i) Expendable \$1000.00 ii) Permanent \$5000.00		

+ principal investigator, ++ research assistant, +++ research associate

(B) - It is estimated that additional useful research in this specific area of noise monitoring could be conducted over an 18 month period with the following breakdown in effort:

(1) Theoretical Modeling	1 ⁺	6 ⁺⁺	0 ⁺⁺⁺
(2) Field Measurements (one or more automated stations)	0 ⁺	5 ⁺⁺	0 ⁺⁺⁺

- (3) Ground/Sky Studies (theoretical work, field measurements and scale modeling) 3⁺ 7⁺⁺ 0⁺⁺⁺
- (4) Equipment i) Expendable \$2000.00
ii) Permanent \$4000.00

ACKNOWLEDGEMENTS

The writers gratefully acknowledge monetary support from both the Council on Environmental Quality and the Graduate School, University of Minnesota, and cooperation from the Public Works Department, City of Minneapolis, and the Pollution Control Agency, State of Minnesota.

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"Elevated Measurement of Urban Traffic Noise"
presented by
R. F. Lambert

DISCUSSION:

L. Faulkner:

How does one incorporate horn blowing into models of city noise?

R. Lambert:

Horn blowing doesn't seem to be quite the problem in the Midwest as it is on the East coast, and we didn't take it into account.

W. Soedel:

Are you saying that the noise level increases with height?

R. Lambert:

Some of the components of noise will increase, but not the total noise.

S. Waggoner:

Why do people in the upper stories complain about noise more than others?

R. Lambert:

Its mainly a psychological effect. You have a different clientele in the upper stories.

K. McConnell:

Were there major highways nearby?

R. Lambert:

Yes.

THE DESIGN OF ENCLOSURES FOR LOW FREQUENCY NOISE CONTROL

G. H. Koopmann, Associate Professor

University of Houston
Department of Mechanical Engineering
Houston, Texas 77004

INTRODUCTION

In the area of low frequency noise control involving interactions between vibrating surfaces and the acoustic pressure fields which they enclose, it is not uncommon to find cases where the structural and acoustical modes exhibit a mutually resonant condition. As a consequence of this condition, high intensity discrete tones can be generated both internally and externally to the enclosing surface, thus causing a serious annoyance or possible hazard to nearby personnel in the noise environment. In the design of cabins for transport vehicles for example, this coupling condition often results in the generation of low frequency interior noise characterized by a series of discrete tones commonly referred to as 'body booms'. The almost predictable reoccurrence of 'body booms' with each new design is ample evidence that, although it is recognized as a structural/acoustical problem, there is still no practical method available to identify which part of the structure couples efficiently with the interior acoustic field. As a result, current corrective measures which attempt to reduce this phenomenon are usually based on a fairly unproductive 'try it and see' basis. Similar problems occur in the design of tight fitting acoustical enclosures. Mutual resonances between the flexible enclosing walls of the cover and the interior annulus of air can greatly reduce and sometimes completely destroy the effectiveness of an enclosure. This is especially true if the structural/acoustical resonance frequency coincides with that of the noise source for which the enclosure was originally intended.

OBJECTIVE OF WORK

The extent of research work reported in this paper covers the general category of structural/acoustical coupling problems described above. These are limited mainly to the low frequency cases where both the structural and acoustical modal densities

are fairly low, and thus a modal approach is feasible. The objective of this work was to develop a method of quantifying the extent of coupling between an acoustic cavity and its enclosing flexible walls at a given frequency in terms of a single number. Such a method would provide the design engineer with a practical means of identifying which surfaces on a complex enclosure are most likely to couple strongly with the interior acoustic cavity. With this information available, it would then be possible to design enclosures with a minimum of modal coupling, and hence a maximum of low frequency noise reduction.

COUPLING MECHANISMS

To examine the nature of the coupling mechanisms involved in enclosure dynamics, it is convenient to divide the discussion into two parts: first, that coupling between structural/acoustical modes which occurs in the frequency domain involving the effects of mutual resonance; and second, that coupling between structural/acoustical mode shapes in the spatial domain involving the matching of modal geometries. Consider coupling in the frequency domain first. The following four cases are possible:

- | | |
|--------|---|
| CASE A | resonant structural/
resonant acoustical |
| CASE B | resonant structural/non-
resonant acoustical |
| CASE C | non-resonant structural/
resonant acoustical |
| CASE D | non-resonant structural/
non-resonant acoustical |

Case A is the most troublesome from the noise control point of view since it represents the strongest coupling condition between the acoustical and structural modal frequencies. The addition of damping to either this structure or the acoustical cavity would act to reduce the coupling to some extent, but at low frequencies this

treatment is only marginally effective. Alternatively, case A could be transposed to the lesser coupled cases B or C or to the least coupled case D by selectively adjusting the modal frequencies of either the structure or the acoustic cavity. This could be achieved by altering the mass or the stiffness of the structure and/or the geometry of the acoustic cavity. When viewed from a practical point of view, however, the method of reducing coupling by effecting modal mismatches in the frequency domain is difficult to implement. Problems of predicting uncoupled modal frequencies of both the structure and the interior acoustical cavity before assembly are formidable. Further, even if a modal frequency mismatch is implemented after assembly, there is always the possibility that any gains achieved could be obviated by subsequent shifts in modal resonances due to changes in ambient temperature or structural boundary conditions over a period of time. This condition would be especially likely in the case of outdoor enclosures covering machines which generate heat as well as sound.

The control of structural/acoustical coupling in the spatial domain bypasses many of the above problems to a large extent. In this domain, the extent of coupling is controlled by adjusting the modal geometries of the structure and its enclosed cavity to ensure that a mismatched condition exists in the frequency bands where coupling is most likely. Since modal geometries are less sensitive to environmental changes than modal frequencies, there is greater certainty that a decoupled condition produced in the spatial domain would remain so over a period of time. It should be noted that constraints which alter modal geometries alter modal frequencies as well, and thus an advantage of treatment in the spatial domain is that alterations to the structure or cavity which produce modal geometry mismatches will produce modal frequency mismatches in addition.

THE JOINT ACCEPTANCE FUNCTION

Quantifying coupling in the spatial domain requires a knowledge of the modal geometries of both the enclosing structure and its interior cavity plus a method of coupling them together. The description of the coupling quantity called the joint acceptance function forms the basis for this paper. The approach will be mainly from an acoustic point of view wherein the structure will be represented in terms of equivalent velocity sources distributed around the boundary of the acoustic cavity. To explain the details of the method, it is useful to consider the simple example shown on

figure 1. In this case, a simply supported flexible plate forms the bottom of a 'hard walled' rectangular box. The question under consideration is, what is the extent of geometric coupling between a set of plate modes (m, n) and the acoustic mode $(1, 1, 0)$ for the condition of mutual resonance at a frequency, f . An answer can be formulated by first writing a Green's function for the Helmholtz equation of the form

$$G\left(\frac{\vec{r}}{\vec{r}_0}\right) = \left[\frac{2\pi}{ipf}\right] \left|\frac{P_s(\vec{r})}{Q_s(\vec{r}_0)}\right| e^{i\theta} P_s, Q_s \quad (1)$$

where $P_s(\vec{r})$ is the acoustic pressure at a field point $\vec{r}$ in the box due to a unit volume velocity source $Q_s(\vec{r}_0)$ at a source point $\vec{r}_0$ on the plate, ρ is the density of the acoustic medium, and θ_{P_s, Q_s} is the phase angle between P_s and Q_s . If the Green's function is known for the surface described by the plate, the field pressure at any point in the box due to an arbitrary velocity distribution of the plate $v(\vec{r}_0)$ can be written as

$$P(\vec{r}) = \oint \left\{ \left|\frac{P_s(\vec{r})}{Q_s(\vec{r}_0)}\right| e^{i\theta} P_s, Q_s \cdot v(\vec{r}_0) \right\} dS \quad (2)$$

Generally, equation (2) offers a means of coupling the motion of an enclosing structure with the interior cavity in the spatial domain. The velocity distribution $v(\vec{r}_0)$ would be given by the mode shape of the enclosing structure within the range of frequencies where coupling in the frequency domain might occur. A more useful coupling quantity based on Equation (2) can be developed as follows. If the arbitrary velocity distribution is chosen such that

$$v(\vec{r}_0) = |Q_s(\vec{r}_0)| e^{-i\theta} P_s, Q_s \quad (3)$$

the corresponding pressure at a given field point would assume a maximum value

$$P_{MAX}(\vec{r}) = \oint \left| \frac{P(\vec{r})}{Q_s(\vec{r}_0)} \right| dA \quad (4)$$

since $|Q_s(\vec{r}_0)|$ has the value of unity. $P_{MAX}(\vec{r})$ would be the total pressure at a given field point obtained by summing over all the unit volume velocities distributed over the surface S_s . The ratio of the field pressure $P(\vec{r})$ due to an arbitrary velocity distribution $v(\vec{r}_0)$ with the above $P_{MAX}(\vec{r})$ conveniently gives the degree of coupling in terms of a single number,

ranging in values from zero (perfect mismatching of modal geometries to unity (perfect matching of modal geometries). The ratio $P(\bar{r})/P_{MAX}(\bar{r})$ can be referred to as the joint acceptance function, J where

$$J = \frac{\left| \int \frac{P(\bar{r})}{Q_s(\bar{r})} v(\bar{r}_s) \cdot ds \right|}{\int \left| \frac{P(\bar{r})}{Q_s(\bar{r})} \right| ds} \quad (5)$$

The expression "joint acceptance" was originally introduced in the analysis of the response of structures to random acoustic loading and gives a measure of the extent to which a particular structural modes 'accepts' a given acoustic trace wavelength. In this work, it can be interpreted in a similar manner, i.e., in terms of modal analysis. Thus, for any arbitrary vibrating surface enclosing an acoustic cavity, the joint acceptance function defines the extent to which the acoustic modes of the cavity geometrically 'accept' the structural modes of an enclosing surface at a given frequency.

USE OF THE J.A.F. - A SIMPLE EXAMPLE

To illustrate how the Joint Acceptance Function can be used at the design stage, consider the following problem. Suppose that two stiffeners were to be added to the plate in the previous example of the rectangular box/plate system. Frequency measurements of the source to be enclosed in the box indicate that the (1,1,0) acoustic cavity mode will be strongly excited. Further, preliminary calculations of the plate response indicate that the (1,2), (2,1), and (2,2) plate modes occur at frequencies at or around the resonant frequency of the (1,1,0) acoustic mode. The question is, where should the stiffeners be placed on the plate to minimize coupling. The J.A.F. shown in figure 1 can be applied to this problem in the following way:

- 1) The geometric coupling between the (1,1,0) acoustic mode and the (1,2) and (2,1) plate modes is weak and thus these modes need not be considered in the question of where to place the stiffeners.
- 2) The geometric coupling between the (1,1,0) acoustic mode and the (2,2) plate mode is very strong. Thus, the stiffeners should not be placed along the two centerlines of the plate parallel to the x and y axis. These locations would only act to enhance the mode and hence the coupling.
- 3) Examination of the geometric coupling in figure 1 as a function of plate modes (m, n) indicates that the two plate modes (1,3) and (3,1) couple weakly with the (1,1,0) acoustic mode.

The stiffeners should thus be placed parallel to one another and either axis at a distance of $L/3$ and $2L/3$ along the length L , of the plate. This placement would suppress the strongly coupled (2,2) plate mode while enhancing a weakly coupled one and hence, minimize the overall structural/acoustical coupling in the frequency range of the enclosed source.

THE J.A.F. - A CASE STUDY

This section presents a few of the results of an investigation on the low frequency interior noise of a lightweight transit van and illustrates how the joint acceptance function can be used as a guide to reducing the interior noise level.

During the measurement of the sound pressure level in the interior of the van, it was observed that several, strongly resonant frequencies occurred in the frequency spectrum as the engine speed was slowly increased from idle to normal operating conditions (approx. 1000-5000 r.p.m.). The peak frequencies along with their amplitudes as measured at the ear of a front seat passenger are given in figure 2. (Note that the pressures are given in dB lin).

To identify the acoustic modes which corresponded to the frequencies of the pressure peaks observed in the full scale experiments, a study on a 1:12 scale model of the van was conducted. Experiments on model verified the existence of the observed acoustic modes and produced a measure of their corresponding shape functions expressed in terms of pressure distributions. With acoustic modal information available, it was then possible to generate the required Green's function $[P_s(\bar{r})/Q_s(\bar{r}_s)]$ over those enclosing surfaces for which coupling was suspected (see next section for details). An example of a Green's function corresponding to the single acoustic mode at 113 Hz is shown in figure 2 under the spatial domain heading.

The next step was to discover which enclosing surfaces of the transit van were coupling strongly with the acoustic space at the troublesome frequencies. Two obvious candidates for this study were the floor and roof panels since both of these were composed of several large, nearly flat, unstiffened surfaces. A typical acceleration response of the floor panel over the frequency range of interest is shown in fig. 2. Note that the resonant peaks of the structural response corresponds closely with those of the acoustic response of the enclosed space. A similar curve was obtained for the structural response of the roof panel. To identify which of the

vibrating surfaces coupled most strongly with the acoustic modes of the space, the velocity distributions $v(\vec{r}_s)$ of the floor and roof panels were measured at each resonant frequency. A typical velocity distribution of the floor at 113 Hz is shown in figure 2. The next step was to couple the measured panel vibrations with the acoustic Green's function at the observed resonant frequencies via equation (5), and compute the corresponding joint acceptance function. In the example shown in figure 2, the computed J.A.F. had a value of 0.83, thus indicating a very strong coupling between the floor mode and the acoustic cavity mode at a frequency of 113 Hz. By performing similar operations on the remaining acoustic and structural response functions, the most strongly coupled surfaces were identified for vibration control treatment.

With identification of the strongly coupled surfaces, the next step was to modify either those surfaces or the acoustic space to produce a decoupled condition. In transport vehicles such as a transit van, modification of the acoustic space is not a practical consideration and thus the response of the enclosing surface must be altered. As a general rule, decoupling is best achieved by producing a geometrical mismatch between the modes of surface and those of the enclosed space. Practically, this can be implemented by attaching stiffeners to the enclosing surfaces to produce a modal response where the antinodal regions of the surface coincide with the nodal regions of the acoustic space and vice versa.

GENERATING GREEN'S FUNCTIONS

In the previous sections it was shown that in order to write the joint acceptance function for a particular system, the dynamic response characteristics of both the structure and enclosed acoustic space must be known. Of the two, the dynamic response of the structure is easier to obtain since measurements on the full scale prototype give exactly the information required in equation (5), i.e. the coupled frequencies and corresponding modal geometries. Full scale measurements of the acoustic Green's function, however, are complicated beyond practicality by the coupled motion of the enclosing structure and thus it becomes necessary to construct, either physically or mathematically, an equivalent model of the uncoupled acoustic space. Two modeling techniques which can be used to generate acoustic Green's functions for irregularly shaped enclosures are described as follows.

One method of acquiring the acoustic response of an irregularly shaped space is to construct a physical model of the space

to a suitable scale. The Green's function $[P_s(\vec{r})/Q_s(\vec{r}_s)]$ can be obtained by exciting the space with a small moveable acoustic source located in the plane of the enclosing surfaces and measuring the corresponding acoustic pressure at a fixed field point, preferably near a corner.

With a suitable model and supporting apparatus available, the modal frequencies of the space can be determined by driving the acoustic source through the frequency range of interest and noting the frequencies at which the pressure at the fixed field point undergoes a maximum response. (During this step care must be taken to avoid placing the source or receiver near the nodal point of an acoustic mode). Following identification of the acoustic resonant frequencies, the next step is to generate an acoustic Green's function $[P_s(\vec{r})/Q_s(\vec{r}_s)]$ for a given surface at each of the resonant frequencies. This is achieved by recording the amplitude and phase of the pressure at a fixed field point while moving the source over those surfaces for which corresponding full scale structural response data are available.

A second means of acquiring an acoustic Green's function for an irregularly shaped enclosure is the acoustic finite element method which necessitates the use of a fairly elaborate computer program. In this mathematical method, the volume of an irregularly shaped space is substituted with a set of smaller, more simply shaped volumes which, in combination, approximate to the original shape. By choosing appropriate shape functions which satisfy the pressure conditions on the boundaries of these smaller volumes, the modal geometry of the original volume can be obtained at each resonant frequency of the cavity. In turn, the Green's function can be generated in terms of a series summation in which each of the terms represent the frequency, geometry, and damping characteristics of each acoustic mode considered. An example of this modelling technique is illustrated in figure 3, where the acoustic space of a transit van is idealized in terms of sixteen finite elements. The corresponding frequencies and shape functions of the first few acoustic modes are also shown (scale 1:12). Note that the slanting of the reflecting surfaces results in modal geometries that are characterized by curved rather than flat nodal surfaces.

CURRENT LEVEL OF ACTIVITY

The method described in this paper is presently being used to study several industrially sponsored projects with a view toward making recommendations for noise

reduction features at the design stage. Theoretical studies on the refinement of the method, particularly with regard to improving the technique of generating Green's functions, are being conducted as a joint project by the author and the following people:

Dr. M. Petyt and J. Lea, I.S.V.R.
University of Southampton, U.K.

Prof. H. Pollard, School of Physics,
University of New South Wales,
Sydney, Australia.

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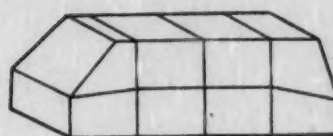


Figure 3(a) Idealization of half van model

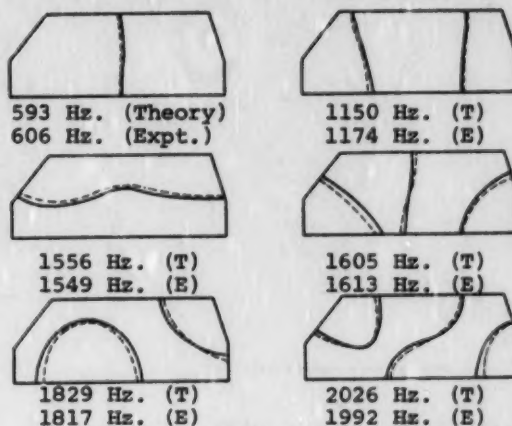


Figure 3(b) Node lines for the acoustic modes of a model van, --- experimental, — theoretical.

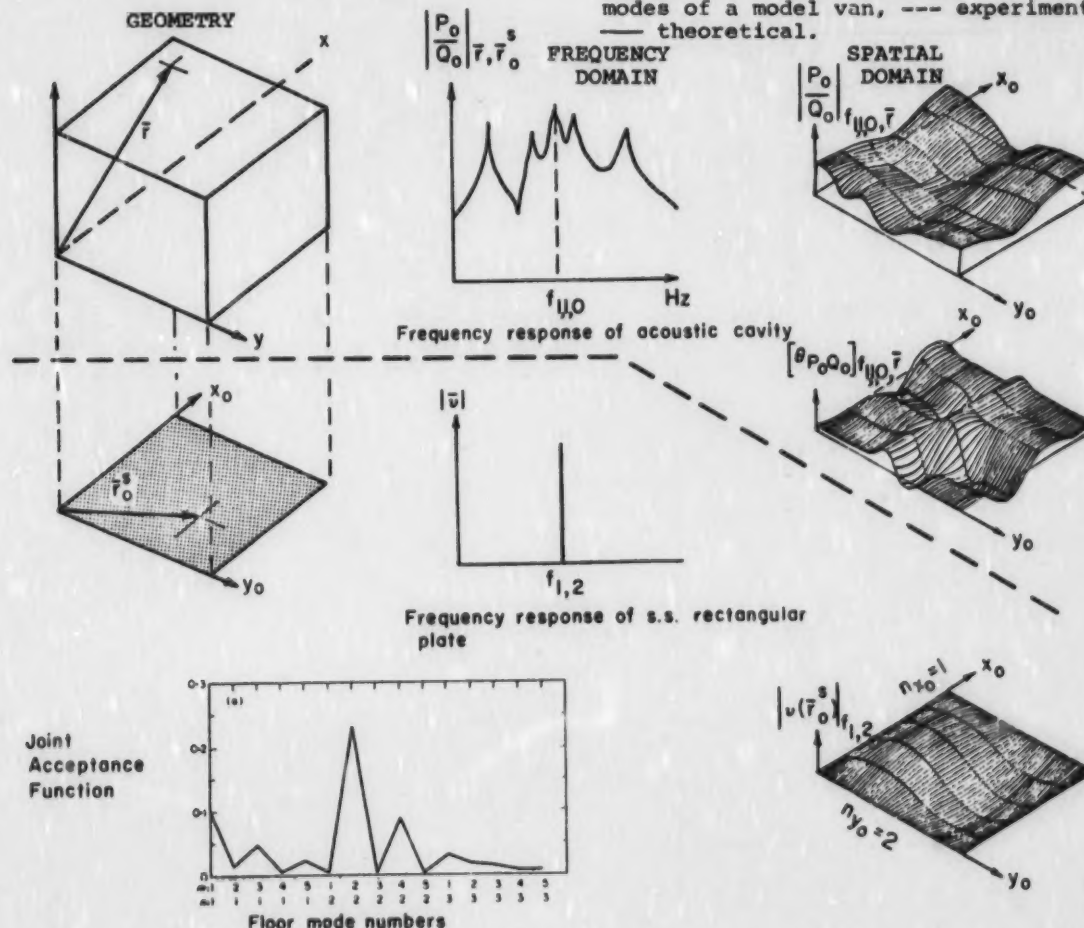


Fig. 1 Joint Acceptance Function for the Coupling of a Hard Walled Acoustic Cavity with a Flexible Bottom

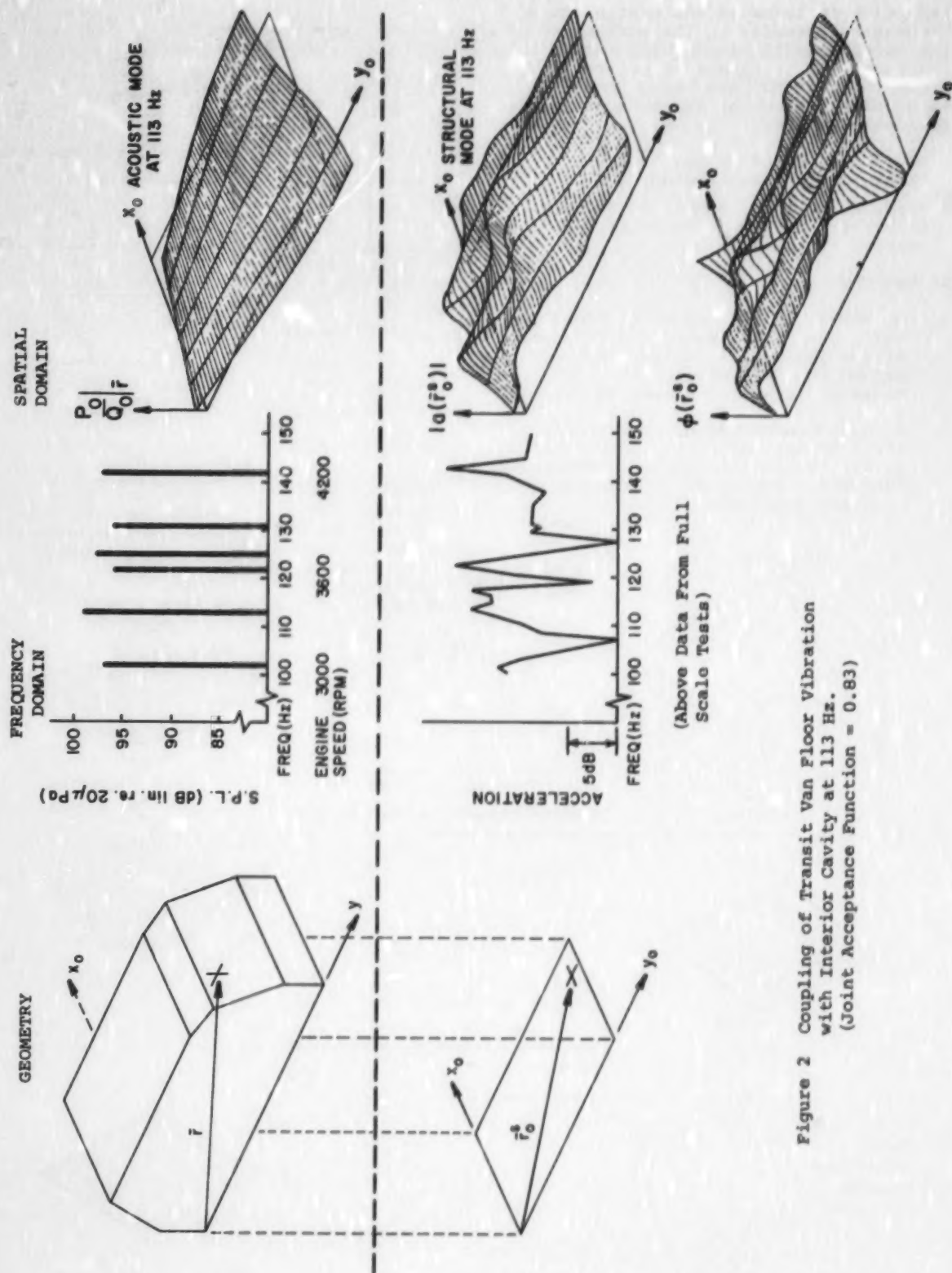


Figure 2 Coupling of Transit Van Floor Vibration with Interior Cavity at 113 Hz.
(Joint Acceptance Function = 0.83)

"The Design of Enclosures for Low Frequency Noise Control"
presented by
G. Koopmann

DISCUSSION:

G. Koopmann:

This project had a funding of about thirty thousand dollars (from industry) covering five to six man years of work.

R. Cohen:

This is a good example of how we can use advanced techniques to solve noise problems.

You mentioned that the low frequency internal noise was about the same whether the van was stationary or rolling. Would not the excitation by the roadway increase the low frequency noise?

G. Koopmann:

In this particular case we found that the excitation by the engine was the major source of noise.

R. Lambert:

This is a good design technique and is much preferable to the type of add-on noise control that is currently being used.

P. Baade:

How did you get the input data for the analysis?

G. Koopmann:

We used both calculated data and experimental data for the input.

W. Soedel:

The biggest problem in implementing an advanced technique like this in industry is getting the industrial people to see its value. They often have the attitude that this is "blue sky" research and has little practical value. The problem is one of education.

DESIGNING DIESEL ENGINES FOR REDUCED NOISE

Richard H. Lyon
Richard G. DeJong*

Massachusetts Institute of Technology
Department of Mechanical Engineering
Cambridge, Massachusetts 02139

APPROACHES TO MACHINERY NOISE REDUCTION

There is an established and widely practiced technology concerned with solving noise problems called "noise control". Although there is no inherent limitation in scope or approach implied by the name, the textbooks and courses dealing with noise control have tended to restrict the meaning to "add-on" procedures for abating noise, such as the use of mufflers, enclosures, isolators, wrappings, and the like. The general implication in noise control is that the noise is dealt with, but the basic design or operation of the device making or transmitting the noise is affected as little as possible.

Noise control so defined tends to be applicable in situations in which the number of installations does not justify the effort involved in redesign, or the technology for such redesign is uncertain. Noise control tends to be expensive per treated unit, and the unit cost does not decrease very much as the number of units increases. The noise control treatment may also interfere with the maintenance, operation, or acceptability of the device.

The alternative to noise control is redesign (for reduced noise). Redesign involves changing the basic construction, layout, or fabrication of the device to reduce the amount of high frequency energy that it emits. Such redesign has a high initial cost -- in the engineering services required to determine the desirable design changes and the retooling necessary to implement them, but the production cost may be insignificantly different from before and one can also institute changes that solve non-noise problems at the same time. Such an approach is usually only available to manufacturers who are able to justify these costs by their sales volume and potential marketing benefits.

The strategy for redesign will be very dependent on the device. One can try to reduce the generation of energy at audible frequencies by modifying impact, combustion processes, or whatever. One can also try to block or otherwise reduce this energy as it propagates from where it is generated to radiating surfaces. Or one can try to keep the energy from being radiated as audible sound. Very often, a mixed strategy will be necessary to obtain the desired degree of abatement.

In our work, we have tended to concentrate on the propagation of energy within the machine, for a number of reasons. First, the modification of the generator of audible energy often affects the other performance parameters in a major way. For example, the high frequency components of combustion chamber pressure are very closely associated with the combustion process itself, and changes in this process have direct interaction with performances, fuel economy, and the production of chemical pollutants. Interference with the process of radiation usually means conventional noise control -- enclosures, wrappings, hoods and the like, although there are procedures such as perforating the exterior surfaces which may be applicable in certain circumstances and can be considered as a part of redesign.

The "path" of vibrational energy within the machine may consist of connecting rods and linkages, rotating shafts and bearings, stiffening frames of various kinds, and the structural framework. Many of these items are installed as manufactured pieces in the machine, so that installing a modified substitute is quite feasible without major changes to the rest of the machine. Thus, path modifications are quite reasonable to contemplate as long as the basic function of the part is also maintained.

PATH ANALYSIS OF A DIESEL ENGINE

We have carried out experimental and theoretical analyses of the transmission of high frequency (audible) energy from the combustion chamber surfaces to the radiating surfaces of the engine casing. The two major paths -- through the head structure and through the piston-connecting-rod-crankshaft assembly -- were evaluated separately. The experimental studies provide us with quantitative information on transfer and input mobility functions, while the theoretical studies allow us to correlate these functional relations with the mechanical configuration of the elements. This latter correlation is of particular importance in evaluating the effect of a design change on noise radiation.

A diagram of the four cylinder diesel engine used in this study is shown in Figure 1. The upper surface of the piston was excited with random noise (with the head in place) and the vibration at a

*now located at Cambridge Collaborative, Cambridge, Mass.

point on the casing determined by carrying out the reciprocal experiment of measuring the vibration of the piston surface when the casing was excited at the measuring location. Then the piston was removed and the head installed and the surface of the head was excited from below, and the casing response measured. The ratio of the velocity of response to drive force is the transfer mobility and is shown for a piston at top dead center.

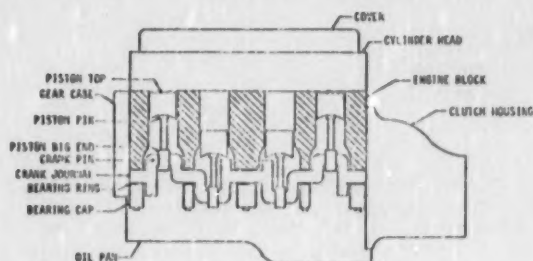


Figure 1: Diagram of Four Cylinder Isuzu Diesel Engine Used in Study Showing Elements of Transmission Paths

The results of the experiment are shown in Figure 2. For equal force amplitudes on the piston and the cylinder head, the piston-connecting rod path to the engine casing will dominate below 2 kHz, while both paths are of nearly equal importance above 2 kHz. For the purposes of this study, we have concentrated our efforts toward understanding the piston-connecting rod paths. We assume that the pistons, connecting rods, and crankshaft are nearly vertically aligned when the combustion pulse is initiated. Consequently, the dynamical model for the elements is for vertical motions and forces, and moments and flexure (except for the crankshaft) are ignored.

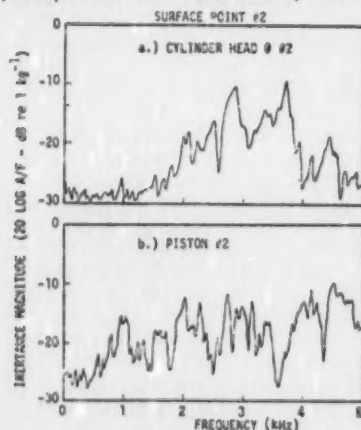


Figure 2: Comparison of Vibration Transmission Through Head-Casing Path and Piston-Connecting-Rod-Crankshaft-Casing Path.

Note dominance of latter path below 2 kHz.

In order to understand this transmission path, we have measured and derived the transfer and input mobilities for the major elements of the path (piston-connecting rod, crankshaft, bearing mount-casing). Experimental consistency between the mobilities of the elements and the calculated mobility of combinations of elements can be very valuable in confirming assumptions about the nature of structural connections. Good theoretical models of the elements, again based on the experimentally determined mobilities of elements, can be quite helpful in suggesting structural modifications that might reduce the radiated noise.

The measured transfer mobility from the top of the piston to the big end is shown in Figure 3, and from the crank pin to adjacent bearing journal of the crankshaft is shown in Figure 4. With the appropriate input mobilities of the elements included, we can combine these measured transfer functions mathematically to predict the transfer function from the piston top to the crankshaft bearing journal. A comparison of the calculated value of this function based on measurements taken on the individual items with a measurement of the overall transfer function is shown in Figure 5. Generally, the agreement is good to about 2 kHz, but it breaks down at higher frequencies. This is probably because the moments at joining points, neglected in the calculation do, in fact, become important at higher frequencies.

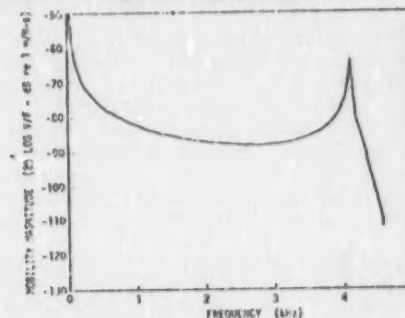


Figure 3: Ratio of Connecting Rod Big-End Velocity to Drive Force at Top of Piston Which is Transfer Mobility Magnitude for the Piston-Connecting Rod Element

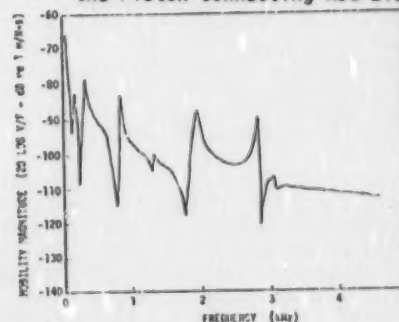


Figure 4: Magnitude of Transfer Mobility From Crankpin (where big end connects) to Crankshaft Journal

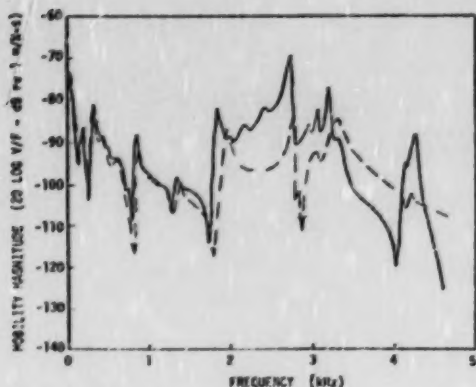


Figure 5: Comparison of Magnitude of Transfer Mobility from Piston Surface to Crankshaft Journal as Calculated from Element Mobilities and by Direct Measurement.

————— calculated -----experimental

As an example of the construction of the model for an element, consider the measured input mobility for the piston-connecting rod element shown in Figure 6. By examining the low frequency behavior of the function, the resonance and antiresonance frequencies, and the magnitude of the mobility at those frequencies, lumped mass-spring representation of the element can be derived, as shown in Figure 7.

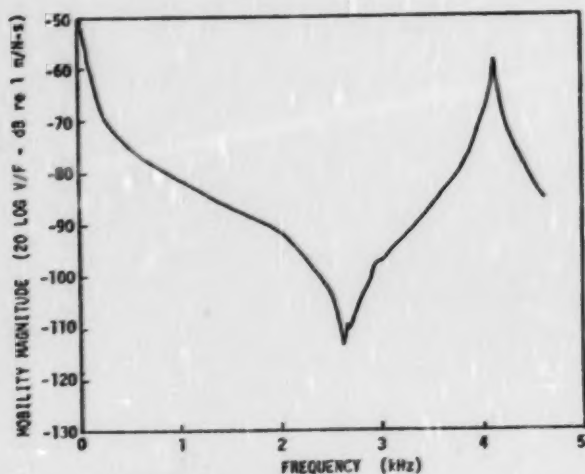


Figure 6: Magnitude and Phase of Drive Point Mobility of Piston-Connecting Rod Element at Piston Surface

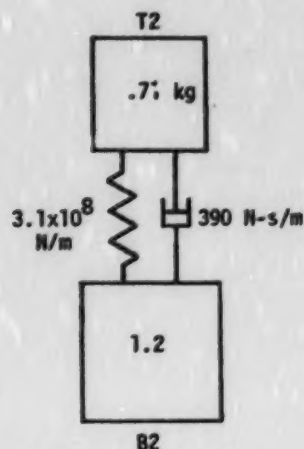


Figure 7: Lumped Mass Model of Piston-Connecting Rod Element Derived from the Data in Figure 6.

We have also developed analytical models based on mass-spring-dashpot elements. These models allow us to predict the effect that changes in the design of the engine on noise radiation. A detailed study of the transfer and input admittance functions reveals that compliance elements located in the bearing crankshaft supports should be quite effective at reducing the transmitted energy at audible frequencies.

To see how the analytical model can help identify useful changes in the structure, consider the diagram of two connecting elements shown in Figure 8. The transfer mobility of the joined system is

$$y_{14}^{(a+b)} = y_{12}^a y_{34}^b / (y_{22}^a + y_{33}^b)$$

We can attempt to reduce $y_{14}^{(a+b)}$ by either decreasing the element transfer mobilities y_{12}^a and y_{34}^b or by increasing one or both of the input mobilities at the junction, y_{22}^a and y_{33}^b . The former option generally requires increasing the mass and stiffness of the system. The latter implies the use of a resilient layer of resonant structure in the junction.

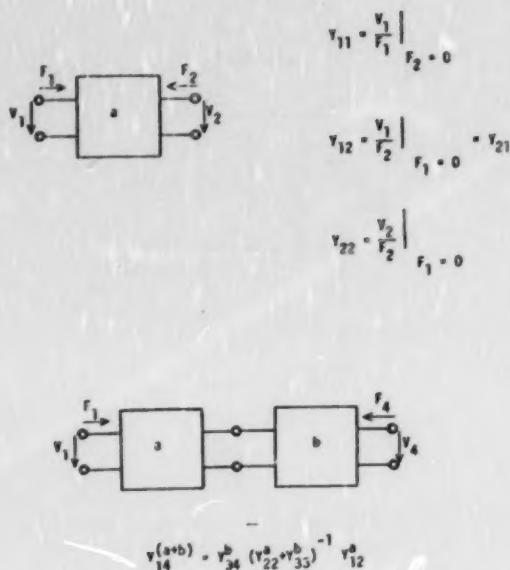


Figure 8: Schematic of Two Mechanical Elements for which the Input and Output Can Be Characterized by a Single Force-Velocity Pair of Variables and Their Associated Mobility Functions

A sketch of a crankshaft bearing ring that incorporates a layer of resilient silicone rubber is shown in Figure 9. Based on our analytical model, the expected reduction in casing radiation that result from the incorporation of such bearings into the engine is shown in Figure 10. This analytical expectation has been confirmed by measurements on the engine structure.

PLANS FOR CONTINUING WORK

Further work on the transmission of high frequency energy within diesel structure will follow two lines--the testing and evaluation of design changes in the environment of an operating diesel engine, and the study of additional design changes for further improvements in the acoustical and non-acoustical aspects of the problem. Clearly, there are many interactions between the design of noise-reduction modifications and the dynamical and other considerations in machine design. Our approach to the problems of design of machines that produce less noise is a total approach, in which we consider the structural and operational implications of design changes. In many cases, the design change desired for reduced noise can have beneficial effects on other machine characteristics (greater structural rigidity, for example).

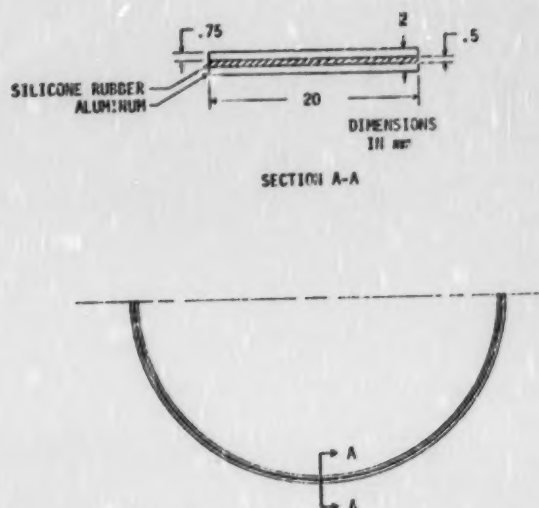


Figure 9: Simplified Diagram of Bearing Ring Modified by Layer of Resilient Silicone Rubber

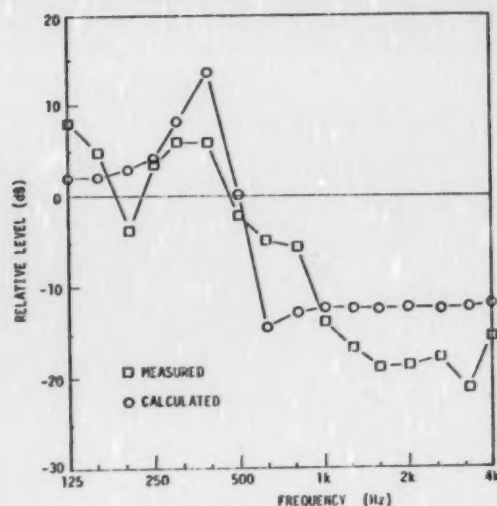


Figure 10: Expected and Observed Changes in Transfer Mobility from Piston Surface to Casing Due to Addition of Resilient Bearing Ring.

Of course, there will be cases in which the desired change (a bearing mounted in rubber, for example) may not be acceptable because of some operational problem such as shaft instability or performance life. Until basic considerations of vibrational performance are included in the machine design strategy, however, we will be forced to live with the unsatisfactory limitations of band-aid noise control.

"Designing Diesel Engines for Reduced Noise"
presented by
R. H. Lyon

DISCUSSION:

R. Lyon:

This work was unfunded and was bootlegged onto other projects. The total level of effort was about two man years.

R. Cohen:

Did you use sinusoidal inputs to get the component characteristics?

R. Lyon:

No, we used random excitation.

R. Cohen:

Were the components removed from the engine when they were tested?

R. Lyon:

Yes.

R. Cohen:

Did you attempt to account for clearances between parts?

R. Lyon:

Yes, when there was an oil film between the parts.

R. Cohen:

Shouldn't there be a natural frequency associated with bending of the connecting rod?

R. Lyon:

Perhaps, but this doesn't show up in

these results. It is detailed in the thesis.

P. Baade:

Can you identify the springs and mass elements in your model as particular parts of the engine?

R. Lyon:

We were more interested in simulating the input power to the system. We didn't want to get involved in a detailed identification of the model parameters.

C. Hurst:

Did you look at fatigue life or stress?

R. Lyon:

No.

J. Sullivan:

Do you expect non-linearities to invalidate your results?

R. Lyon:

Perhaps.

STUDIES OF COMBUSTION AND MECHANICALLY INDUCED NOISE IN DIESEL ENGINES

Andrew F. Seybert
Department of Mechanical Engineering
University of Kentucky
Lexington, Kentucky 40506

INTRODUCTION

Studies have shown that surface transportation (trucks, buses, cars) is the major cause of noise complaints in most industrialized countries [1]. In the United States at least twice as many people are disturbed by traffic noise as aircraft noise [2]. It is estimated [2] that surface transportation exposes between 15 and 33 million people to an outdoor Day/Night Sound Level, L_{dn} , exceeding 65 dB, a level that has been found to produce widespread complaints as well as threats of legal action [3].

Of the more than 125 million vehicles in the United States only about 3.75 million are classified as medium or heavy duty trucks (Gross Vehicle Weight Rating exceeding 10000 lb.). These trucks only comprise about 10% of the traffic on major routes [1] but are responsible for about 75% of the total sound power radiated by all vehicles [2].

The most annoying aspect of traffic noise is the fluctuating noise level resulting from the fact that vehicular traffic is intermittent. For typical traffic flows of automobiles only, these fluctuating noise levels are not noticeable if one is farther than about 200 feet from the roadway. However, since trucks are about 10 dBA noisier than cars, and are more widely dispersed, the fluctuating noise levels are extended an order of magnitude farther from the roadway [1].

Under the mandates of the Noise Control Act of 1972, the U.S. Environmental Protection Agency has identified trucks as a major noise source in the United States [4] and has issued noise limit regulations for trucks already engaged in interstate commerce. In addition, EPA has set noise limits for new trucks. Effective January 1, 1978 the maximum permissible noise level will be 83 dBA at 50 ft, and will be

reduced to 80 dBA, effective January 1, 1982.

DIESEL ENGINE NOISE

The diesel engine is usually one of the three most important noise sources of most heavy duty trucks [5]. The other major noise sources are usually (but not limited to) the cooling fan and the engine exhaust. Noise control hardware (better mufflers, more efficient fans) is generally available to reduce the cooling fan and engine exhaust noise. Once this has been done the diesel engine is normally the major source of noise. This, coupled with the fact that 96% of new heavy duty trucks are diesel-powered [2], requires that the mechanism of diesel engine noise production be understood if significant gains are to be achieved in truck noise control.

Traditionally, diesel engine noise has been subdivided into combustion induced and mechanically induced noise (note: intake and exhaust noise are not considered as "diesel engine" noise, but are categorized separately). Combustion noise is noise induced by the fluctuating gas forces within the combustion chambers of the engine. The dominant transmission paths for combustion noise are:

1. The cylinder head
2. the cylinder liner and water jacket region
3. the piston/connecting rod/crankshaft mechanism.

The transmission of energy along these paths is dependent on the dynamic properties of the engine structure, particularly the stiffness and damping. Attempts to show the importance of the structural response of the engine block, and its effect on noise, have resulted in exotic engine structures such as the "skeleton" engine (engine-block walls were replaced by a

highly damped material [6]), the "bearing beam" engine (a thick beam was used to connect the main bearing caps together [7]) and the "magnesium" engine [8]. Except for the magnesium engine, which resulted in a noise reduction of about 10 dBA (it had a wall thickness of 1 inch), increasing the structural stiffness by redesign appears to yield only about a 3 dBA reduction of noise [7].

Considerable insight into the mechanism of combustion noise is gained by studying the combustion pressure time history. Figure 1 shows three typical cylinder pressure time histories for three engine conditions (2400 rpm, 7, 60 and 120 ft-lb torque) for a V-type naturally-aspirated, direct-injection diesel engine, rated at 155 bhp at 3300 rpm, Figure 2.

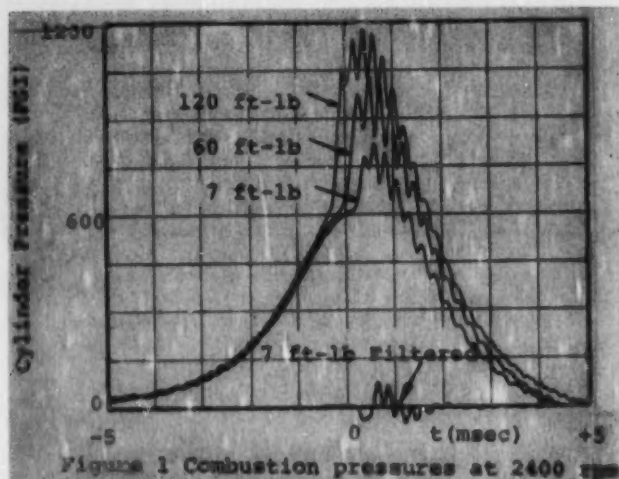


Figure 1 Combustion pressures at 2400 rpm



Figure 2 Diesel engine in anechoic chamber.

The most important aspect of these curves, as far as noise is concerned, is the rapid pressure rise at the beginning of combustion. When fuel is injected into the combustion chamber, there is a short time delay while the fuel vaporizes, spreads throughout the chamber and is heated to the combustion temperature. Once the appropriate temperature and pressure are reached, ignition takes place. However, this initial phase of combustion is accompanied by a rapid rise in pressure due to the burning of the excess fuel accumulated in the chamber during the ignition delay period. After this initial phase of combustion, the fuel burns more slowly and the pressure reaches a maximum slightly after top-dead-center (TDC), then decreases during the expansion stroke. The high rate of pressure rise just after ignition is generally proportional to the ignition delay and causes the characteristic diesel engine "knock".

Superimposed on each of the pressure time histories is a high frequency oscillation due to pressure wave resonance in the combustion chamber. This combustion chamber mode is of negligible importance to noise since it is of very high frequency.

The small fluctuating curve at the bottom of Figure 1 is what remains when the 2400 rpm, 7 ft-lb pressure time history is filtered to remove the pressure components below 800 Hz. This is quite striking when one considers that most diesel engine noise is above 800 Hz. As will be discussed later, the engine structure does not radiate sound efficiently at either very low or very high frequencies, and it is the mid-frequency components (800-3000 Hz) of the combustion pressure that excite the noise producing modes of the engine block.

More detailed aspects of the combustion noise phenomenon are revealed by examining the frequency spectra of the combustion pressure time histories. Figure 3 shows the pressure spectra (bandwidth = 20 Hz) for the three histories in Figure 1 where the portion of the spectrum below 800 Hz has been removed before the spectra were computed. (Below 800 Hz the spectra increase monotonically.) In the 1 to 3 kHz region the increase in spectrum level with torque is due to the more rapid pressure rise for higher torques. The rate of pressure rise for the three curves in Figure 1 is: 7 ft-lb - 1200 psi/msec, 60 ft-lb - 1500 psi/msec, 120 ft-lb - 2000 psi/msec.

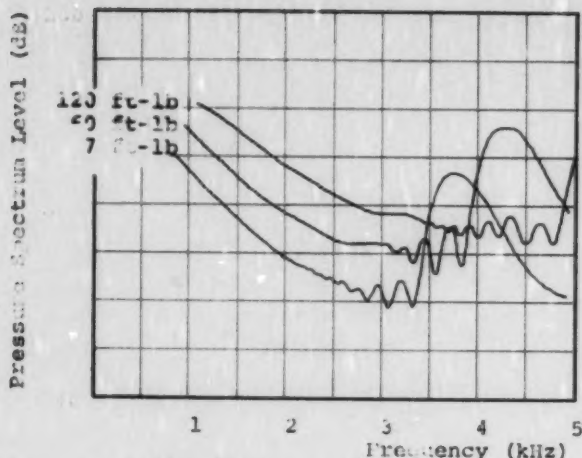


Figure 3 Combustion pressure spectra at 2400 rpm.

Higher rates of pressure rise result in increased excitation of the engine block structure in the critical mid-frequency range, resulting in increased combustion noise.

The broad peak above 3500 Hz in each of the spectra is the combustion chamber resonance seen in Figure 1. The frequency of resonance increases for higher torques since the gas temperature (and the speed of sound) is higher.

Figure 4 is a spectrum of the 7 ft-lb torque pressure history with a bandwidth of 10 Hz. The individual harmonics of the combustion pressure are clearly revealed by using the smaller bandwidth. For the engine speed of 2400 rpm the fundamental firing frequency is 20 Hz and the higher harmonics are integer multiples of 20 Hz. Figure 4 also reveals some interesting aspects of the combustion chamber resonance, namely, it is a broad-band random phenomenon. This is probably due to the fact that the gas temperatures and the combustion chamber volume are changing during combustion, causing the frequency of resonance to change (a careful study of the oscillations superimposed on the curves in Figure 1 shows that the frequency of resonance decreases during the expansion stroke).

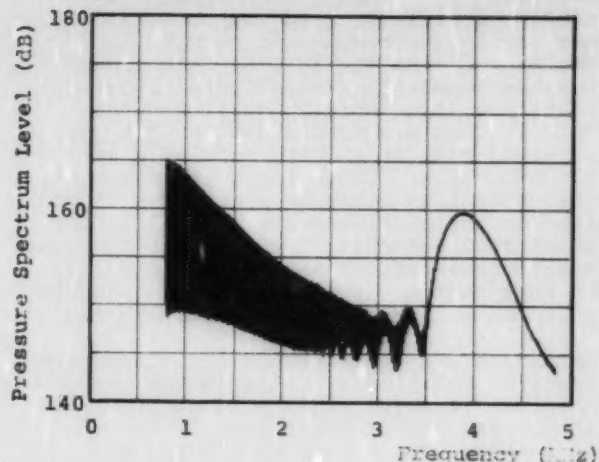


Figure 4 Combustion pressure spectrum at 2400 rpm, 7 ft-lb torque.

In summary, one concludes that in the low and mid-frequency range the combustion pressure is dominated by energy concentrated at multiples of the firing frequency, and at high frequency the combustion pressure is essentially a broad-band random process. Figure 5 shows the variation of overall engine noise level with speed and torque where the microphone was located approximately 1m from the side of the engine. The increased noise with speed is due to the additional excitation caused by a shift of the combustion spectra to higher frequency as a result of increased speed.

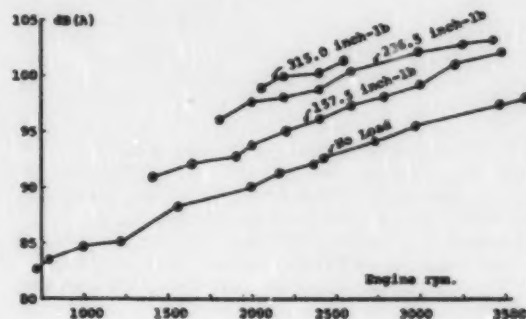


Figure 5 Variation of A-weighted sound level with engine speed and torque.

The most significant reductions in combustion noise of four-cycle direct-injection diesel engines have been obtained by retarding the ignition timing and turbocharging [9]. When a turbocharger is used, the pressure (and temperature) of the air in the combustion chamber is higher and when the fuel is injected it reaches combustion temperature sooner, resulting in a shorter ignition delay. This in turn produces a smaller rate of pressure rise during the initial phase of combustion. Thus, for a given torque, the combustion pressure spectrum will be less in the mid-frequency range than for a naturally-aspirated engine with the same torque, even though the peak pressure will be much higher. Consequently, the combustion noise is reduced by turbocharging.

Retarding the ignition timing reduces the ignition delay in a similar manner since the fuel is injected at a later time when the pressure and temperature of the air in the combustion chamber are higher. Turbocharging and retarding the timing have the additional benefits of reducing smoke and certain gaseous emissions, however, efficiency may decrease in some cases.

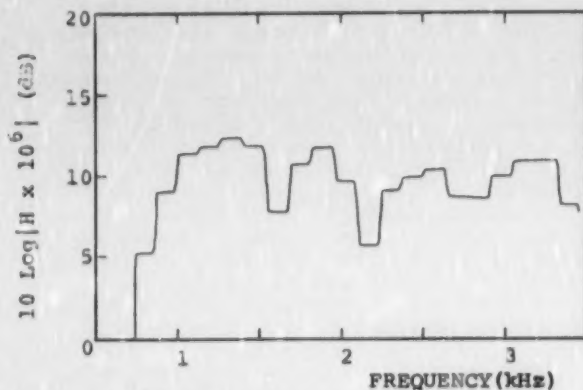


Figure 6 Frequency response magnitude H.

Figure 6 shows a typical frequency response (bandwidth = 140 Hz) for the V-type engine under study [10]. Regions where the frequency response is high correspond to high dynamic response and/or high radiation efficiency of the engine block. Consequently, where the frequency response is high the engine noise will also be high. Below 500 Hz and above 3500 Hz the frequency response is quite low, resulting in negligible noise even though the combustion pressure spectrum is high in these frequency ranges.

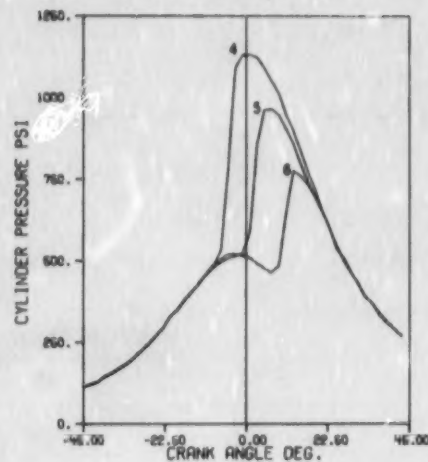


Figure 7 Typical Cylinder pressures

Using measured values of frequency response for the engine structure, as shown in Figure 6, one can predict the effect of combustion modifications on noise, providing the combustion pressure time histories are known. Figure 7 shows three combustion pressure time histories of equal torque but with different ignition timing. Using these curves, and knowing the frequency response of the engine structure, the combustion noise was predicted for each of the three ignition timings [10], and is shown in Figure 8.

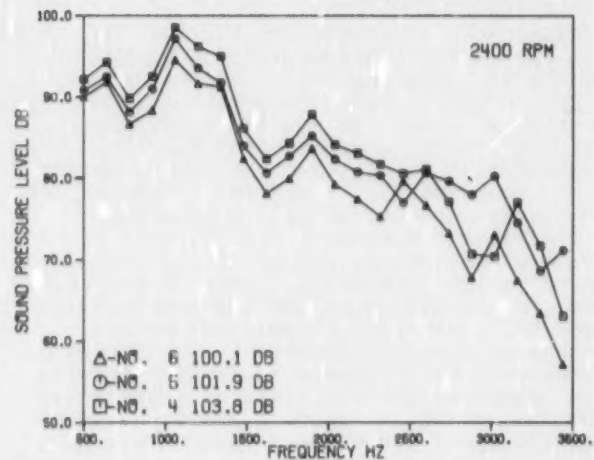


Figure 8 Computed Sound Pressure Levels

MECHANICALLY INDUCED NOISE

Mechanically induced noise in diesel engines is caused by mechanical impacts (due to clearances), friction, and rotating imbalance. The mechanical noise is usually determined by motoring the engine and noting the noise level. The combustion noise is the difference between the motored noise and the total noise. This procedure assumes that the mechanical noise is independent of the combustion pressures. This assumption may be adequate for some mechanical noise sources such as valve train noise, but is in error for other mechanical noise sources such as bearing and piston impact noise, where the mechanical impacts are highly dependent on the magnitude of the combustion pressures. To measure more accurately the noise contribution of a mechanical noise source, one can effectively "eliminate" the source by an appropriate modification, and note the difference in noise level. For example, the noise contribution of piston impacts is often quantified by substituting pistons of low clearance for standard pistons.

PISTON SLAP NOISE

Turbocharged engines are usually dominated by mechanical noise sources, and the major mechanical noise source is usually due to piston impacts [9]. A piston impact occurs when the piston transfers from the non-thrust to the thrust side of the cylinder liner just after TDC during the combustion phase of the engine cycle, Figure 9. Piston impacts occur at several other times during the engine cycle but only the impact at TDC during combustion is significant, since the combustion pressures acting on the piston at this point cause the impact to be quite severe.

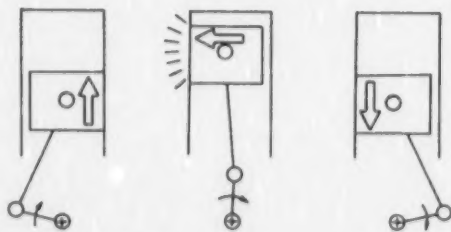


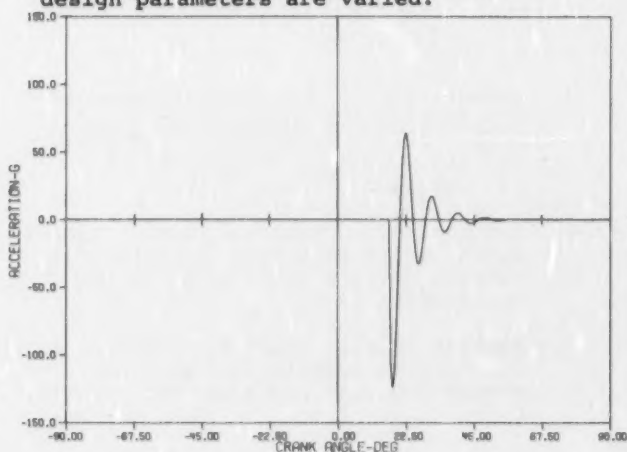
Figure 9 Piston impact sequence

Figure 10 Response of cylinder liner to piston impact.

Piston impacts can be reduced by: using oversized pistons to reduce the running clearance, displacing the piston/connecting-rod pin to one side of the longitudinal axis of the piston (this causes two smaller impacts when the piston hits the liner instead of a single large one), or by using a shim (made of teflon-impregnated sintered bronze [9], for example) which is attached to the side of the piston and holds the piston on one side of the cylinder liner throughout the engine cycle. All of these modifications have drawbacks and usually reduce the noise only 1 or 2 dBA.

Work currently under progress is directed at predicting the response of the cylinder liner to piston impact as a function of liner and piston geometry, running clearance, and combustion pressure characteristics (pressure time history, timing, etc.). It is reasonable to assume that piston impact noise is proportional to the vibration of the cylinder liner (there is some experimental evidence to confirm this [11]) and a mathematical model predicting liner vibration can be used to estimate the relative effect on noise of changing any of the parameters in the model.

The model first calculates the piston side-force for a given geometry and combustion pressure time history. The liner is modeled as a simple oscillator and the response to the piston side-force is calculated. The piston momentum at impact is also calculated and used in the calculation of the liner response. Figure 10 is a typical liner response predicted by the mathematical model. Experimental data will be used as a verification of the piston impact model. Once the model is refined, design charts will be constructed to show the relative effect on noise when key design parameters are varied.



SUMMARY

The mechanisms of combustion and mechanically induced diesel engine noise are being investigated in an effort to reduce the noise from heavy duty trucks. It was shown that by examining the combustion pressure spectrum and the engine structural frequency response that many of the characteristics of combustion noise could be explained. In particular, it was shown that naturally-aspirated diesel engines (direct-injection) are usually dominated by combustion noise in the 500-3500 Hz region since both the frequency response and the combustion pressure spectrum are high in this region, as compared to turbocharged engines of equivalent speed and torque, where the combustion pressure spectrum is lower.

Piston impact noise, a major source of noise on turbocharged engines, is being simulated mathematically with the aim of developing design guidelines to aid engineers in making practical design modifications to reduce this source of noise.

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"Studies of Combustion and Mechanically
Induced Noise in Diesel Engines"
presented by
A. F. Seybert

DISCUSSION:

A. Seybert:

This work was initially funded by the National Science Foundation and is now being funded by industry. The total funding over the past five years has been about fifty thousand dollars. The total level of effort has been approximately three to four man years.

APPENDIX 1
SUMMARY DISCUSSION

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OPEN DISCUSSION ON SOCIETAL NEEDS IN NOISE CONTROL*

Co-Chairmen: R. Cohen (Purdue) and Harvey Nozick (EPA)

H. Nozick:

Speaking for EPA, I want to personally thank each of you for participating in this seminar. I thought it was extremely worthwhile and I hope that you share that opinion. I think the benefits of it will be seen once the documentation gets released and gets widespread circulation.

As a regulatory agency we must consider the available state-of-the-art technology, in addition to other things, when we promulgate regulations for the protection of the public health. In cases where there is an inadequate data base to help us form these regulatory actions, we may go out and fund programs to demonstrate the state-of-the-art for the technology, either unilaterally or in conjunction with another agency. We have a couple of these demonstration programs ongoing at the present time.

To give you a frame of reference of the kinds of actions that we are considering, most of you probably know that we've already promulgated regulations on heavy trucks and compressors. We are in the final stages of preparing notices on six other products: motorcycles, buses, loaders and dozers, truck mounted compactors and refrigerator units. These are pretty far along in our regulatory process at the moment. What I wanted to emphasize more now are the other products that we are considering for future actions, and any help that this group can give in providing some additional technology for the data base in these areas would be well received. Let me just highlight them: automobiles, light trucks, tires, snowmobiles, rapid rail transit systems, pavement breakers and rockdrills, earthmoving equipment, lawnmowers, air conditioners, chain saws, and a variety of home and consumer products. Some of these are in the very early stages of consideration. If you have some suggestions or ideas, we will be happy to listen to them. Anything you can provide to us will lead to a

better regulation in the long run. I would now like to sit back and listen to what you have to offer.

R. Cohen:

Both government and industry are interested in noise control. The university is an ideal environment in which to explore noise control methodology and practical noise control solutions. In particular, universities have the qualified people and necessary facilities that small and medium sized industries lack. Interaction between government, university and industry helps educate students who will in turn enter industry upon graduation. Most industries find that they can support noise control research on the university level much cheaper than they can internally. These are all important reasons why industry chooses to support noise control research at universities. An additional benefit to industry as a result of university sponsored research is the contact that their engineers have with experts in the field. This is a type of continuing education for practicing engineers.

Many of you have submitted items for discussion at today's meeting. We will spend roughly half of the remaining time in discussion of these issues. The remaining time will be spent discussing other ideas that anyone might have.

Several of you commented that you would like to see more fundamental research supported at the university, as opposed to research on a specific product. Does anyone wish to comment?

Messrs. Tichy, Hixson and Hodgson commented that there seems to be no government agency interested in funding basic research in noise control. Noise control research "falls in the cracks" of a number of government agencies, and is not "general" enough for consideration by the National Science

*Editor's note: The comments have been abridged and edited for conciseness and clarity by the Editors. We have endeavored to convey the original intention of the speakers, and we accept the responsibility of any misinterpretation on our part. In one particular case (Bailey, N.C. State) a prepared statement was read. This statement has been reproduced with only minor editing.

Foundation. The National Science Foundation, in particular, just doesn't seem interested in basic research in airborne acoustics. Tichy also mentioned the need for standardization of noise measurements.

H. Nozick:

As John (Schettino) said yesterday, when EPA was created and the Noise Control Act was passed, there was no intention of giving EPA another charter to do research where that capability existed in other areas. The NBS and NSF do have the charter to do that. We have just recently reactivated federal interagency research panels. The function of the panels is to assess the ongoing research and development that is being funded by the federal government. Hopefully, we can identify needs or voids in the funding arena and through this medium initiate federal action, not necessarily done by EPA but done by the agency that has the proper capability.

P. Baade (Purdue):

I cannot resist this opportunity to pick up the key words spoken by Jiri (Tichy)--standardization and measurement methodology. I think that there is no question that noise control in the United States has received a tremendous boost through the Noise Control Act passed by Congress and the responsibility given to EPA to pass regulations for noise emission standards and, ultimately, product noise labeling. As one does this, the problems of measurement technique and accuracy become important. In the international arena there are a number of standards that have come out just recently, some of them still in draft, but already very advanced, and need only to be ratified by the member bodies. In the United States, we do not have the counterparts for these measurements standards. It is futile, in the long run, to regulate noise emission unless there is a sound measurement basis in a coherent structure that covers all types of noise sources. The basic standards for the physical measurement of sound have not been completely updated in the last fifteen or twenty years.

Several persons mentioned the problem of inadequate time to respond to EPA published RFP's. The problem seems to be one of getting the RFP through university channels in time for an interested faculty member to respond adequately.

R. Lyon (MIT):

Concerning the problem of responding to RFP's, I believe many of the problems encountered here are related to the poor timing from an academic year point of view and the limited time of work. It is not always possible to complete a master's degree program in a specified length of time, say 9 or 12 months.

I am also concerned that some university research is of a very mundane nature with little educational value. This work should be done by commercial organizations.

R. Cohen:

Bill Gatley wants to comment on the need for education in noise control.

W. Gatley (Missouri-Rolla):

I think there is a national need for education in noise control, and this education should begin in the public school system. Even at the university level, only a small percentage of the students receive formal education in noise control.

Most engineering curricula are already overcrowded and it is almost impossible to add a mandatory course in noise control. The same is true of elementary and secondary education curricula. The alternative is to introduce noise control in other related courses; for example, in a course on machine design.

H. Nozick:

Consumer education in noise control is very important. We are aggressively moving in a consumer education program. We now are using an exhibit which is being sent around to high density areas to inform people on the effects of noise, what can be done, etc. We are also developing several bulletins and handouts.

D. Tree (Purdue):

I am personally aware of one public school system that includes a section on noise in their grade school science class; however, what is taught is presented incorrectly.

We also have abuses in noise control education. Once every semester, I give a guest lecture on noise to a class in another department on campus. A few of these students have then claimed that they have "experience" in noise control. This type of situation is appalling and needs to be corrected.

D. Muster (Univ. of Houston):

The Acoustical Society (of America) has the Coordinating Committee for Environmental Acoustics. These people have information available for lectures on the subject of noise control and acoustics. There are regional coordinators who have lists of people available for giving talks at community meetings.

D. Tree:

The American Society for Engineering Education also has an Acoustics Committee.

H. Nozick:

The aviation industry has an impressive education program. McDonnell Douglas has developed a simplistic slide presentation on noise and the effects of noise related to aviation. In addition, AIAA has put out several films and slide/tape shows on noise.

R. Lambert (Minnesota):

I wanted to describe one of the things that we have done at Minnesota, which I think has been very effective. Three of us, a mechanical engineering professor in air pollution, a civil engineering professor in solid waste, and an electrical engineering professor in noise offer a course available to students in the college of liberal arts. We call it "Introduction to Environmental Technology" and we typically get from seventy-five to one hundred students per year who take this course. We talk to them about the problems involved, including the potential limitations of technology in solving these problems. The responses that we obtained from counselors on liberal education have been very favorable to this course.

P. Baade:

With regard to education in noise control, I believe that the Institute of Noise Control Engineering has a commitment to this education, but this commitment is limited, primarily, to the practicing engineer. We do this through annual conferences and our bi-monthly publication, Noise Control Engineering.

R. Cohen:

Another problem area that has been mentioned is one of providing accessibility and dissemination of noise control solutions to industry when the research is proprietary in nature.

Messrs. Muster and Lyon commented that it is natural for companies to want to protect the results of university research sponsored by them, particularly if the project is very hardware oriented.

R. Cohen:

When negotiating a research contract with industry, agreement should be reached on what material can be published and what cannot. I find that this avoids problems concerning confidentiality. However, if the work must be kept completely secret, then it doesn't belong at a university.

Professor Hurst has a comment regarding industry/government relationships.

C. Hurst (V.P.I.):

Wherever possible EPA (and other regulatory agencies) should seek to avoid an adversary relationship with industry. Perhaps joint funding ventures between government and industry might help the adversary problem.

H. Nozick:

We are considering just that possibility. You have to remember, though, that EPA has a mandate in the Noise Control Act that says EPA will protect the public health and welfare. Sometimes, that just is an adversary position to making a buck.

W. Soedel (Purdue):

Industrial sponsors of noise control projects often do not allow progress beyond a point of fixed achievement. More basic research has to be bootlegged, as a fact of life. Purely fixed noise control is not research that is valuable in the long range, and we should not rely on it completely for academic problems. Coupled with the practical problem of noise control should be research that aims to uncover fundamental aspects that apply to a variety of devices, or classes of machinery. Money should be made available to supplement industrial contracts to allow follow-up studies of fundamental questions.

It is unfair to industry for the government to regulate the noise from its products and not to provide technical help in the form of education.

F. Hart (N.C. State):

We can help industry by making available qualified people to help solve industry's noise control problems. We do this mainly through research grants that serve as M.S. and Ph.D. theses.

D. Muster:

The problem is very acute among small industries (3 to 4 million dollars per year sales). In a company of this size a single engineer may have a wide range of responsibilities, and cannot be an expert in noise control. We need to make available to these people information on noise control.

With regard to education, I think we should be more careful with nomenclature and terminology. During the technical sessions I noted many terms and phrases that were misused or were not even defined.

R. Cohen:

Frank Hart would like to comment on cooperative programs.

F. Hart:

University/industry/government cooperative programs are important. Right now we need a number of demonstration programs to show what can be achieved in the area of noise control using existing technology.

D. Muster:

There must be a way to get practical information on noise control to practicing engineers who do not have time to read the journals. Perhaps EPA can help these people in this regard.

C. Hurst:

It would be very difficult to find a product that is manufactured in this country that does not have a manufacturer's association associated with it. These organizations could be used for distribution of noise control information to their member companies.

F. Hart:

One comment about the trade associations. We went out to a number of plants and did studies in different situations, developed the data, did some pilot studies on eliminating some of the difficult problems and brought this data back in and held regional seminars throughout the U.S. We have had good response to this type of approach and I recommend it highly.

W. Roper (EPA):

I wanted to comment just a little bit about this area of stimulating some type of cooperation between the regulated or potentially regulated industry and the agency, in this case EPA, that's in the process of developing regulations. I'm personally placed in that position every day, and the studies that we have ongoing that are in direct support of regulation development put us in a situation where we have to look at conflicting interests when we award the contract. We are legally limited or not allowed to let contracts directly with the industry that is going to be potentially regulated because there is a direct conflict of interest that eliminates that area of cooperation. I think there are some other things that can be done, and one is to keep the industry apprised of our progress. When our contractors come in with draft reports, we review them in-house and send these out to knowledgeable people in industry to get their comments. We've also held meetings with industry representatives to discuss specific issues that are critical to a particular rule-making. This past July we held a meeting on Measurement Methodology for Automobiles and Light Trucks in which all the principle industry and environmental groups had a chance to participate. However, we are legally limited, once we begin the regulatory development process, to what we can and can't do from a cooperative standpoint; we cannot contract directly with industry. Of course, we can contract with universities and other unbiased sources.

J. Bailey (N.C. State):

One thing which I would like to comment on is what I see as a national need. That is, some focusing on the problem of occupational exposure to noise. It's been my observation that those most affected by occupational noise have the least ability to deal with the situation. I'm speaking about those citizens who are daily insulted, abused by noise in their job. For many years we've known that excessive exposure to noise causes permanent noise induced threshold shift or deafness, yet what are we doing? It's been distressing to observe how little progress has been made. One only needs to review testimony at recent public hearings on the issuance of a noise standard to understand the present state of affairs.

One problem which I believe impedes progress is a lack of incentives. Unfortunately, due to present circumstances, many of the organizations which are big enough to have capabilities in

noise control are also the ones which tend to have large capital investments in noisy equipment. Do they have real incentives to demonstrate noise control? Do they have any incentives to publish and distribute or, in any way, communicate their results? On the other hand, smaller companies don't have a capability for noise control, regardless of whether or not they have any incentives. I think that's one problem, and historically government has stepped in to meet problems like this.

If we are going to do anything but stick acoustic band-aids on our problems, I think we must recognize the need for research. A first step might be to define some specific problems in occupational noise exposure. In other words, a list similar to the one which you presented, Harvey (Nozick), of the problems that EPA has identified. I don't know of such a list in the area of occupational noise. With or without such a list, though, I think it is apparent that we are dealing with a problem of national importance and it will require a large, continuous effort if we are going to get anywhere.

I see these problems as being of two types. The first type is existing problems where noise levels are perhaps marginally above acceptable levels; there is a tremendous need for trained individuals to handle these problems. Let me give you an example. The textile industry has about 7,000 plants; if we try to take our N.C. State research team into each of these plants it would take us more than 20 years to spend one day in each plant (if we work 7 days a week). I think there is a role for universities in addressing this particular problem. One method which immediately comes to mind is through short courses. Our department has had several hundred individuals for such short courses over the past three or four years, and perhaps that will help.

The second type of problem is one which I think can be characterized as one requiring a major breakthrough. For example, inherent noise from certain kinds of operations. We perhaps are beginning to understand them but we don't see very many solutions coming forth at this point.

I'd like to reinforce some of the things that Ray (Cohen) said. I think universities are uniquely positioned to lead the effort here. We are accustomed by training and experience to be open-minded in our approach to problems like this. We are well equipped with instrumentation, computers, and personnel. Our primary function is to teach, and surely we have a role in training both engineers as well as other people with less train-

ing in the area of noise control. Perhaps most importantly, universities traditionally do research in the public domain. I don't think I need to discuss any further the problems associated with proprietary information and the good that it does. I don't know what business universities could ever have in that area if we consider education to be our main function.

Another point I would like to make is that I believe universities can be very cost effective. Let me give you an example. We've been supported for a three year study by the National Institute for Occupational Safety and Health. We have an effort of about a dozen man years, and we've spent more than \$300,000. We provided information to reduce noise exposure for approximately one half of the million or so workers in the textile industry to 90 dBA or less. Cost effectiveness: let me give you one example using techniques remarkably similar to those Professor Lyon talked about today for a diesel engine. We've attacked a major problem, that of textile spinning, modelling it piece by piece, putting the pieces together, using all the technique that we know how to use. We've demonstrated on approximately 1000 spindles, in 3 plants in 2 states, that noise levels can be reduced to below 90 dBA at the cost of \$2-7 a spindle. Now this compares with latest industry estimates given earlier this month of \$31-51 a spindle. Since there are 19 million spindles, one can calculate an enormous potential savings for the textile industry which, of course, is eventually a saving to consumers. At the same time, and perhaps more importantly, we've eliminated an intolerable health hazard for thousands of people. At least we have provided the information.

Now, let me describe roles that I see for the university and government in attempting to address some of these problems. The first category, I think we have a duty, an obligation, a moral responsibility to communicate simple techniques to those individuals who do not have technical backgrounds but nevertheless have responsibilities for solving existing noise problems. The second category of problems will require good basic research. Surely, universities and governments have some responsibility to participate in this.

Specifically, let me get to the problems of support which have already been mentioned. We've all experienced frustrations in trying to initiate programs to solve some of the problems which have been described. Is it really in the national interest for so much confusion to exist, as to who has responsibilities for what? Surely, the technical problems

are difficult enough for us.

The final point, which has already been made, I suppose, is that it is expensive to educate a specialist in noise control. Almost every graduate student requires support, financial support, in some way. And, surely, this is a problem of sufficient national interest that the federal government can provide some input into this.

E. Sneckenberger (W.V.U.):

I would like to mention two problems that I have encountered with respect to funding of noise control research. The first is a problem with research funding from industry. After a researcher solves the immediate need of a company, there is little incentive for the company to fund additional research to investigate more fundamental questions about the problem.

The other problem concerns the difficulty that those of us who do not have large laboratories have in getting research funding in noise control. We can't possibly go after the big contracts so our only alternative is to act as a subcontractor to some other research organization. Perhaps there is something that can be done to better utilize the expertise of the smaller university research labs around the country.

K. McConnell (Iowa State):

This problem of lack of facilities is a great penalty to smaller universities around the country. We are told, in effect, that we can't do the research because we don't have the equipment, but we can't get the equipment without the research. It's a vicious cycle.

I would like to see EPA investigate noise exposure in small towns. Almost every small town has a grain elevator, with grain drying capabilities, and the noise levels can be quite annoying to the residents nearby. Most of the companies that manufacture this equipment are low technology companies that do not have the knowledge (or the incentive) to apply existing noise control developments to their products.

R. Cohen:

Gentlemen, thank you for coming and for your frank comments and pertinent presentations. The Seminar is over and the last session closed.

APPENDIX 2
ROSTER OF ATTENDEES

EPA-UNIVERSITY NOISE SEMINAR

ROSTER

October 18-20, 1976
Purdue University
West Lafayette, Indiana 47907

BAADE, P.K., Visit. Prof., Purdue University, 171 Brookside Lane, Fayetteville, NY 13066
BAILEY, J.R., N.C. State University, P.O. Box 5801, Raleigh, NC 27607
BARRY, K., Noise Regulation Reporter, Bureau of National Affairs, 1231 25th street, NW,
Washington, D.C. 20037
BRACH, R.M., Assoc. Prof. University of Notre Dame, Aerospace & Mechanical Engr.,
Notre Dame, Indiana 46556
CHANG, Y.M., Engineer (Acoustics), EPA Noise Enforcement Facility, P.O. Box 2089,
Sandusky, Ohio 44870
COHEN, R., Director, Herrick Laboratories, School of Mechanical Engineering, Purdue
University, West Lafayette, Indiana 47907
ELY, F., Project Officer, EPA, Washington, D.C. 20460
FAULKNER, L.L., Assoc. Prof., Ohio State University, 206 W. 18th Ave., Columbus,
Ohio 43210
FONTAINE, W.E., Herrick Laboratories, School of Mechanical Engineering, Purdue
University, West Lafayette, Indiana 47907
GATLEY, W.S., Prof. of Mechanical Engineering, Aerospace, University of Missouri,
1111 Sycamore Dr., Rolla, Missouri 65401
GIBSON, J.D., Prof. of Mechanical Engineering, Rose-Hulman Inst. of Tech., Terre Haute,
Indiana 47803
GREENE, E.R., Asst. Prof. Univ. of Alabama - Birmingham, School of Engr.,
Univ. Station, Birmingham, Alabama 35294
HAMILTON, J.F., Prof. Mech. Engr., Purdue University, West Lafayette, Indiana 47907
HART, F.D., Prof., North Carolina State University, Center for Acoustical Studies,
Box 5801, Raleigh, North Carolina 27607
HIXSON, E., Prof., University of Texas, Austin, Texas 78712
HODGSON, T.H., North Carolina State University, Center for Acoustical Studies,
P.O. Box 5801, Raleigh, North Carolina 27607
HOLGER, D.K., Asst. Prof., Iowa State University, 103 Lab of Mechanical Dept. of
Engr. Sci., & Mech., Ames, Iowa 50011
HURST, C.J., Assoc. Prof. of Mech. Engr., Virginia Polytechnic Inst. & State Univ.,
Dept. of Mechanical Engineering, Blacksburg, Virginia 24061
KOOPMANN, G., Assoc. Prof., University of Houston, Dept. of Mechanical Engineering,
Houston, Texas 77004
LAMBERT, R.F., Prof., University of Minnesota, Elec. Engr. Dept., Minneapolis,
Minnesota 55455
LYON, R.H., Prof., MIT, 77 Mass Ave., Rm. 3-366, Cambridge, Massachusetts 02139
MANSBACH, P.A., Physicist, National Bureau of Standards, Washington, D.C. 20234
MC CONNELL, K.G., Prof., Engr. Science & Mechanics, Iowa State University,
205 Lab of Mechanics, Iowa 50010
MC DANIEL, O.H., Pennsylvania State University, 213 Engr. Unit E, University Park,
Pennsylvania 16802
MC KINNERY, W.N., Chief, Physical Agents Control Section, NIOSH, 4676 Columbia
Parkway, Cincinnati, Ohio 45226
MUSTER, D., Brown & Root Prof., University of Houston, Dept. of Mech. Engr.,
Houston, Texas 77004

NOZICK, H.J., Acting Chief - Technology, EPA, Washington, D.C. 20460

RAJ, D., Grad. Instructor, Mechanical Engr. Dept., Tennessee Tech. Univ.,
Cookeville, Tennessee 38501

ROPER, W., Branch Chief, Surface Transportation, EPA, Washington, D.C. 20460

SCHETTINO, J.C., Dir. Tech & Fed. Prog. Div., EPA, Washington, D.C. 20460

SEMRAU, W.R., Project Engr., Frito-Lay Inc., P.O. Box 2231, Irving, Texas 75060

SEYBERT, A.F., Asst. Prof., University of Kentucky, Lexington, Kentucky 40506

SNECKENBERGER, J.E., Assoc. Prof., West Virginia University, Mechanical Engineering
and Mechanics, Morgantown, West Virginia 26505

SOEDEL, W., Prof. of Mechanical Engineering, Purdue University, West Lafayette,
Indiana 47907

SULLIVAN, J.W., Asst. Prof., Mechanical Engineering, Purdue University, West
Lafayette, Indiana 47907

SWIM, W.B., Prof. School of Mechanical Engineering Tennessee Technological
University, Box 5014 TTU, Cookeville, Tennessee 38501

TICHY, J., Prof., Penn State University, Applied Research Lab, University Park,
Pennsylvania 16802

TREE, D.R., Prof., Ray W. Herrick Laboratories, Purdue University, West Lafayette,
Indiana 47907

WAGGONER, S., Development Engineer-Cannery Noise, University of California, Ag. Engr.
Dept., UCD, Davis, California 95616

APPENDIX 3
COMPENDIUM OF UNIVERSITY RESEARCH IN NOISE CONTROL

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COMPENDIUM OF UNIVERSITY RESEARCH IN NOISE CONTROL

The following compilation is a result of a questionnaire sent to over 300 colleges and universities in the United States requesting information on status of current and recently completed noise control projects. Although the main purpose was to determine the extent of funding of noise control research by the private sector, projects with other sources of funds are also listed. Each university is listed in alphabetical order along with the mailing address and the department in which the research was conducted.

The tabulated information includes: a short, descriptive title of the project; the amount of funding, either total or annual; the source of funding, either industry, university, or government agency; the level of effort in man months, where the three columns represent the level of effort by the principal investigator(s), graduate student(s) and research associate(s) respectively; the principal investigator(s); and the actual or estimated completion date. To be included in the compendium, research projects had to satisfy the criteria that they have funding exceeding \$1000 and/or represent a total level of effort exceeding two man months. Only projects in progress or completed in the past three years are listed.

PROJECT TITLE	AMOUNT OF FUNDING	SOURCE	LEVEL OF EFFORT (MAN-MOS.)	PRINCIPAL INVESTIGATOR(S)	COMPLETION DATE
AUBURN UNIVERSITY DEPARTMENT OF INDUSTRIAL ENGINEERING AUBURN, ALABAMA					
Guard Design to Reduce Cir- cular Saw Blade Noise	\$ 2,000	Univ.	2 0 0	Trucks	1/77
UNIVERSITY OF CALIFORNIA-DAVIS DEPARTMENT OF MECHANICAL ENGINEERING DAVIS, CALIFORNIA 95616					
Silencing of Two-Stroke Motorcycles	\$ 72,000	Indus.		Karnoff	1/76
<u>Department of Agricultural Engineering</u>					
Cannery Noise Research	\$ 7,000	Univ.	1 2 6	Roley	1974
Reduction & Control of Noise Levels in Canneries	\$ 44,500	Indus.	1 20 12	Waggoner	3/76
Reduction and Control of Noise Levels in Canneries	\$ 37,600	Indus.	1 20 12	Waggoner	5/77
UNIVERSITY OF CINCINNATI MECHANICAL ENGINEERING DEPARTMENT MAIL LOC. 72 CINCINNATI, OHIO 45221					
Noise Levels of Control Valves	\$ 8,700	Indus.	8 1 3	English & Brown	1973
Line Silencer Performance	\$ 5,000	Indus.	6 4 2	English	1974
Garbage Truck Noise	\$ 1,000	Indus.	0.5 1 0	Morse	1974
Transient Sound Propagation	None		2 6 0	Morse	1977
BEST-EAR	None		2 6 0	Morse/Brown	1977
CORNELL UNIVERSITY DEPARTMENT OF MECHANICAL AND AEROSPACE ENGINEERING 105 UPSON HALL ITHACA, NEW YORK 14853					
Helicopter Rotor Noise	\$ 30,000/yr	ARO	3 24 0	George	

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PROJECT TITLE	AMOUNT OF FUNDING	SOURCE	LEVEL OF EFFORT (MAN-MOS.)	PRINCIPAL INVESTIGATOR(S)	COMPLETION DATE
FLORIDA ATLANTIC UNIVERSITY DEPARTMENT OF OCEAN ENGINEERING BOCA RATON, FLORIDA					
State Needed Research in Urban Noise Control	\$ 30,000	FL DER	3 18 0	Dunn	10/77
State Needed Research in Urban Noise Control	\$ 30,000	FL DER	3 18 0	Dunn	10/76
FHWA Noise Model Evaluation Data Project	\$ 20,000	FL, DOT & FHWA	3 6 0	Dunn	3/76
State needed Research in Urban Noise Control	\$ 10,000	FL DER	1 6 0	Dunn	1/75
Noise Model Evaluation Project	\$ 11,000	FL DOT	3 6 0	Dunn	1/74
FLORIDA TECHNOLOGICAL UNIVERSITY MECHANICAL ENGINEERING AND AEROSPACE SCIENCES DEPARTMENT P.O. Box 25000 ORLANDO, FLORIDA 32816					
Noise Program to cope with Central Florida Regional Commu- nity Noise Problems	\$ 19,500	FL DER & SUS	4 6 0	Ventre	12/76
GEORGIA INSTITUTE OF TECHNOLOGY SCHOOL OF AEROSPACE ENGINEERING ATLANTA, GEORGIA 30332					
Combustion and Entropy Noise in Turbopropul- sion Systems	\$ 36,000/yr	NASA	2 4 6	Strahle	7/77
Random and Un- steady Combustion Processes in Diesel Engine Noise	\$ 52,000/yr	NSF	2 4 7	Strahle	12/76
Acoustic Proper- ties of Turbo- fan Inlets	\$45,000/3yrs	NASA Lewis	0.6 6 2.4	Zinn	8/77
UNIVERSITY OF ILLINOIS AGRICULTURAL ENGINEERING DEPARTMENT URBANA, ILLINOIS 61801					
Noise and Vi- bration of Off- Road Equipment	\$ 30,000 annually	Ill. Agr. Expt. Sta. & Industry	4 24 0	Yoerger	8/77

PROJECT TITLE	AMOUNT OF FUNDING	SOURCE	LEVEL OF EFFORT (MAN-MOS,)	PRINCIPAL INVESTIGATOR(S)	COMPLETION DATE
IOWA STATE UNIVERSITY MECHANICAL ENGINEERING DEPARTMENT AMES, IOWA 50011					
Multistage Axial-Flow Turbomachine Wake Produc- tion, Trans- port and Inter- action	\$168,000 (to date)	AFOSR, ISU, NSF	6 6	Okishi	funded till 9/30/77
<u>Engineering Science and Mechanics Department</u>					
Crop Dryer Noise	\$ 10,000	ISU	2 2 0	Holger	6/76
Agricultural Fan Noise	\$ 10,000	Indus.	2 1 0	Holger	10/75
Power Offset Sensitivity Study	\$ 7,000	USAF,CEC	3 0 0	Holger	8/76
UNIVERSITY OF KANSAS DEPARTMENT OF AEROSPACE ENGINEERING 2004 LEARNED LAWRENCE, KANSAS 66045					
A Research Pro- gram to Reduce Interior Noise in General Aviation Airplanes	\$ 29,914	NASA -Langley	3 15 3	Roskam	1/77
UNIVERSITY OF KENTUCKY FORESTRY DEPARTMENT LEXINGTON, KENTUCKY 40503					
Noise Control in Woodworking Industry	\$ 90,000	USDA & KY State Gov- ernment	0.3 0.1 0	Tang	1978
THE UNIVERSITY OF MASSACHUSETTS MECHANICAL ENGINEERING DEPARTMENT AMHERST, MASSACHUSETTS 01002					
Wide Noise Barriers	Unfunded	None	2 3 0	Ivey	6/77
Acoustic Enclo- sures	Unfunded	None	2 3 0	Russell	8/77
MASSACHUSETTS INSTITUTE OF TECHNOLOGY DEPARTMENT OF MECHANICAL ENGINEERING 3-360 CAMBRIDGE, MASSACHUSETTS 02139					
Response Distri- bution in Broad- Band Random Vibration	\$ 72,000	NSF	5 24 18	Crandall	
Acoustical Modeling for Site Evaluation	\$147,800	NSF/RANN	6 24 0	Lyon	
Res. on Noise Propagation	\$151,481	NSF	6 48 12	Lyon	4/75

PROJECT TITLE	AMOUNT OF FUNDING	SOURCE	LEVEL OF EFFORT (MAN-MOS.)	PRINCIPAL INVESTIGATOR(S)	COMPLETION DATE
UNIVERSITY OF MINNESOTA ELECTRICAL ENGINEERING DEPARTMENT MINNEAPOLIS, MINNESOTA					
Noise Monitor- ing	\$ 11,000	Sloan Fdtn. University	1 6 0	Lambert	6/76
Flow Duct Acoustics	\$ 60,000	NSF	10 24 0	Lambert	12/78
Spatial Filter- ing Techniques	\$ 25,000	NSF	4 12 0	Lambert	9/74
MISSISSIPPI STATE UNIVERSITY DEPARTMENT OF AEROPHYSICS AND AEROSPACE ENGINEERING DRAWER A OR DRAWER AP MISSISSIPPI STATE, MISSISSIPPI 39762					
SRB Heat Shield Fluctuating Pressure Tests	\$ 20,000	NASA	4 6 0	Shepard	12/76
UNIVERSITY OF MISSOURI-ROLLA DEPARTMENT OF MECHANICAL & AEROSPACE ENGINEERING ROLLA, MISSOURI 65401					
Jet Impingement Noise Reduction through Surface Acoustical Treatment	\$ 5,000	Univ.	2 12 0	Oetting	1/76
A Study of the Structural- Acoustic Res- ponse & Interior Noise Levels of Fuselage Structures	\$ 29,606	NASA Lang- ley Research Center	6 12 0	Koval	5/77
Resources for Education in Acoustics	\$ 3,000	MO Industri- al Commission Halliburton Co.	3 0 0	Gatley	12/76
THE UNIVERSITY OF NEBRASKA COLLEGE OF ENGINEERING AND TECHNOLOGY LINCOLN, NEBRASKA 68588					
Agricultural Noise Exposure Research Pro- gram	\$ 14,000	FIEI, NIOSH	12 0 0	Splinter	12/77
<u>Department of Engineering Mechanics</u>					
Suburban Noise Control Using Combinations of Trees, etc.	\$ 20,000	USDA FS	9 0 0	Cook	1/77
<u>Agricultural Engineering Department Tractor Test Lab</u>					
Sound Measure- ment on Tractors	Part of Test fee	Indus.	0 0 5	Mumgaard	
Computer Stor- age of Noise Level Measure- ments and their Analysis	\$ 1,000	Tractor Test Lab	0 0 3	Leviticus	

PROJECT TITLE	AMOUNT OF FUNDING	SOURCE	LEVEL OF EFFORT (MAN-MOS.)	PRINCIPAL INVESTIGATOR(S)	COMPLETION DATE
THE UNIVERSITY OF NEBRASKA - CONTINUED					
<u>Mechanical Engineering Department</u>					
Control of Punch Press Noise Thru Variable Resilient Mounting	\$ 20,700	Indus.	6 12 0	Baumgarten	7/75
NEW JERSEY INSTITUTE OF TECHNOLOGY DEPARTMENT OF MECHANICAL ENGINEERING NEWARK, NEW JERSEY 07102					
Noise Transmis- sion through Building Wall	\$ 20,000	Univ.	6 0 0	Nahavandi, Sun, Ball	1976-77
Fluid-Solid Interaction	\$ 20,000	Univ.	6 12 0	Nahavandi	1976-77
NORTH CAROLINA STATE UNIVERSITY CENTER FOR ACOUSTICAL STUDIES BOX 5801 RALEIGH, NORTH CAROLINA					
Coordinated Textile Indus- try Noise Re- duction Program	\$383,869	NIOSH	36 60 36	Bailey & Emerson	4/77
Punch Press Noise Reduction	\$ 53,815	NIOSH	12 15 0	Bailey	9/75
Noise Control for General Aviation Air- craft	\$ 58,000	NASA	2 24 0	Hart & Bailey	6/77
Metallforming Machinery Noise Analysis	\$ 8,857	Indus.	1 5 0	Bailey	1/74
Lighting System Noise	\$ 5,314	Indus.	1 6 0	Bailey	6/74
Wood Planer Noise	\$ 10,000	Indus.	6 12 0	Hart	6/73
Noise Test Code	\$ 8,000	Indus.	3 6 0	Hart	1/73
Circular Saw Noise Analysis	\$ 5,000	Indus.	2 12 0	Reiter	6/74
Acoustics Training Program	\$ 80,000	NASA	4 36 0	Hart	8/75
Noise Control of Wood Planers	\$100,000	NIOSH	12 24 0	Hart	9/76
Woodworking Machinery Noise	\$ 20,000	Indus.	3 15 0	Hart	8/73
Tire Noise	\$250,000	DOT	2 48 12	Reiter	6/77
Combustion Noise	\$ 50,000	DOT	6 24 0	Hassan	12/76
Industry Noise Control	\$ 90,000	NC-DOL	12 0 0	Royster	6/77

PROJECT TITLE	AMOUNT OF FUNDING	SOURCE	LEVEL OF EFFORT (MAN-MOS.)	PRINCIPAL INVESTIGATOR(S)	COMPLETION DATE
NORTH CAROLINA STATE UNIVERSITY - CONTINUED					
Noise Problem Identification and Solution Synthesis	\$ 60,000	Indus.	7 24 0	Hart	8/73
Aerodynamic Diffusion	\$ 10,000	NASA	2 0 9	Hart	8/73
Transportation Noise Symposium	\$ 5,000	DOT	2 0 0	Reiter	8/74
Acoustic Radia- tion from Ro- tating Disk	None	None	12 36 0	Hodgson	11/76
Transient Sound Generation at Impact	\$ 7,000	Internal	12 36 0	Hodgson	12/75
Noise Measure- ment on large machines in situ	\$ 10,000	New York State Sci. & Tech. Foundation	12 72 0	Hodgson	5/76
UNIVERSITY OF NORTH DAKOTA DEPARTMENT OF MECHANICAL ENGINEERING GRAND FORKS, NORTH DAKOTA 58202					
Noise Reduction in IC Engine Exhaust Systems	\$ 2,000	Indus.	2 15 0	Sadler	5/77
Noise Survey of a Lignite Pro- cess Develop- ment Unit	None		1 1 0	Sadler	5/77
THE OHIO STATE UNIVERSITY DEPARTMENT OF MECHANICAL ENGINEERING 206 W. 18th AVE. COLUMBUS, OHIO 43210					
Lawnmower Noise Reduction	\$ 14,000	Indus.	2 6 0	Faulkner	6/74
Basic Study of Impact Noise	\$ 10,000	Indus.	3 6 0	Faulkner	9/74
Impact Machine Noise Study	\$ 8,000	Indus.	2 6 0	Faulkner	6/75
Interaction of Sound, Flame Stability and Products of Combustion	\$ 40,000	Indus.	8 16 0	Faulkner	9/75
Study of Sound Attenuators	\$ 12,000	Indus.	3 6 0	Jones	6/76
Noise Reduction by Barriers Indoors	\$ 4,000	Univ.	0 6 0	Moran	9/76
Stamping Mach- ine Noise	\$ 4,000	Indus.	1 6 0	Faulkner	9/75
Transmission Loss of Pipe Wrapping System	\$ 6,000	Indus.	2 6 0	Faulkner	12/75

PROJECT TITLE	AMOUNT OF FUNDING	SOURCE	LEVEL OF EFFORT (MAN-MOS.)	PRINCIPAL INVESTIGATOR(S)	COMPLETION DATE
OKLAHOMA STATE UNIVERSITY SCHOOL OF ARCHITECTURE STILLWATER, OKLAHOMA 74074					
Visual Contri- bution to Speech Perception in Noise	\$ 5,000	Univ.	4 1 0	Boyer	6/76
THE PENNSYLVANIA STATE UNIVERSITY APPLIED RESEARCH LAB P.O. BOX 30 STATE COLLEGE, PENNSYLVANIA 16801					
Axial Flow Fan Stage Unsteady Performance	\$ 33,000	Navy	5 6 0	Bruce	9/77
Scattering From Edges	\$ 34,500	Navy	0 18 0	Hayek	12/76
Transient Noise Radiation	\$ 40,300	Navy	0 24 0	Hayek & Stuart	6/77
Near-Field of Shells	\$ 43,600	Navy	0 18 0	Hayek	6/77
Platelike Dyna- mic Absorbers	\$ 39,000	Navy	0 18 0	Snowdon	6/77
Investigation of Unsteady Pressure Distribution on Axial Flow Fan Blading	\$ 25,000	Navy, NASA	1 8 0	Henderson	9/77
The Unsteady Response of an Axial Flow Fan to Inlet Distor- tions	\$ 18,000	NASA	1 6 0	Henderson	9/77
Propulsor Radi- ated Noise	\$185,000	Navy	2 0 12	Lumley & Henderson	9/77
Interaction of Inlet Turbulence and Guide Vane Secondary Flow with Propulsors	\$ 35,500	Navy	3 6 0	Thompson & Lakshminarayana	9/77
Cavitation and Noise Scaling	\$100,000	Navy	2 0 9	Holl & Henderson	9/77
Far-Field Sound Sources on Sim- plified Propel- ler Blades	\$ 25,000	Navy	1 6 0	Lauchle	9/77
Noise Studies Associated With Axisymmetric Turbulent Boun- dary Layer Flow	\$ 50,000	Navy	5 0 0	Lauchle	10/77
Active Control of Sound Fields	\$ 23,700	Navy	0 11 0	Lawther	9/77
Hearing Damage	\$ 13,250	Navy	0 8 0	Michael & Prout	9/77
Helmoltz Inte- gral Study	\$ 9,400	Navy	0 5 0	Thompson & Penlon	9/77

PROJECT TITLE	AMOUNT OF FUNDING	SOURCE	LEVEL OF EFFORT (MAN-MOS.)	PRINCIPAL INVESTIGATOR(S)	COMPLETION DATE
THE PENNSYLVANIA STATE UNIVERSITY - CONTINUED					
Effects of Gas Inclusions on Sound Propaga- tion in Visco- elastic Materials	\$ 23,500	Navy	0 13 0	Ricker	9/77
Study of Some Reflection and Diffraction Phenomena	\$ 33,700	Navy	0 19 0	Skudrzyk	9/77
Vibration and Sound of Plate- like Structures	\$ 14,000	Navy	0 8 0	Skudrzyk	9/77
Engine Vibration Isolation Pro- gram	\$200,000	Navy	20 12 24	Ricker	9/77
Aerospace Engineering Department 233 Hammond Building					
Jet Noise & Coherence Struc- ture	\$ 46,000	NASA	3.6 18 0	Arndt	
Cavitation Noise	\$ 10,000	Netherlands Navy	3 0 3	Arndt	
Pressure Field in a Turbulent Jet	\$ 12,000/yr	Navy	1.2 6 0	Arndt	
Aeroacoustic Research in Wind Tunnels	\$ 30,000	NASA	2.4 6 0	Arndt	1973
Department of Mechanical Engineering 213 Engineering Unit E					
USDA Acoustics	\$ 25,789	USDA For.	2 12 6	Reethof	7/75
NASA Acoustics V	\$ 52,632	NASA	3 12 12	Reethof	7/74
NSF Duct Acoustics	\$ 57,052	NSF	3 12 12	Reethof	9/73
NASA Acoustics IV	\$ 26,316	NASA	2 24 0	Reethof	3/76
USDA Acoustics	\$ 27,333	USDA For.	2 12 6	Reethof	9/76
NASA Acoustics Trainee	\$ 25,273	NASA	0 24 0	Reethof	8/74
NASA Duct Acoustics	\$ 26,316	NASA	1 12 12	Reethof	12/75
NSF Noise Control	\$ 52,842	NSF	3 12 12	Reethof	12/75
Valve Mfgs. Collaboration	\$ 5,000	Indus.	0 6 0	Reethof	4/77
NSF Noise Control	\$ 52,400	NSF	3 12 12	Reethof	11/76
PRINCETON UNIVERSITY AEROSPACE AND MECHANICAL SCIENCES DEPARTMENT PRINCETON, NEW JERSEY					
Interior Acous- tics	\$ 40,000/yr	NASA	1 2 0	Dowell	continuing

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PRINCETON UNIVERSITY - CONTINUED					
Space Shuttle Structural Response	\$ 40,000/yr	NASA	1 1 0	Dowell	11/76
PURDUE UNIVERSITY SCHOOL MECHANICAL ENGINEERING WEST LAFAYETTE, INDIANA 47907					
Piston Impact Noise	\$ 10,000	Indus.	2 6 0	Seybert	8/77
Lawn Mower Tractor Noise	\$ 86,500	Indus./Univ.	13 31 0	Tree & Crocker	12/76
Noise Source Identification & Muffler Design for Garden Tractors	\$ 43,336	Indus.	8 20 0	Tree & Soedel	12/76
Rotary Compressor Noise	\$ 20,000	Indus.	4 6 0	Tree	8/76
Noise & Vibration From Transporta- tion Vehicles & Other Machinery	\$300,000	NSF		Crocker, Cohen, Tree, Sullivan	12/76
Noise Source Identification on Trucks & Reduction of Noise by Total Engine Enclosure	\$159,000	EPA	10 24 0	Tree, Sullivan, Crocker	12/77
Comp. Shell Dyn.	None		1/2 3 0	Hamilton	6/77
Plate Damping Coatings	\$ 25,000	Indus.	1 1/2 6 0	Hamilton	6/74
Duct Acoustics By Finite Elements	None		1 1/2 6 0	Hamilton	8/76
Piston Compressor Noise Study	\$ 30,000	Indus.	3 7 0	Tree	8/77
New Muffler Concepts	\$ 33,000	Indus.	3 8 0	Sullivan	12/76
Automotive Muffler Design	\$ 75,000	Indus.	5 24 0	Crocker, Sullivan	8/77
Truck Noise Measurement	\$130,000	Indus.	9 27 0	Sullivan	12/77
Noise Control of Chain Saws	\$ 30,000	Indus.	14 14 0	Soedel	3/76
Gas Pulsations in Four Cylinder Automotive Comp.	\$ 40,000	Indus.	15 15 0	Soedel	5/75
Gas Pulsations in Refrigeration Compressor Pipes	\$ 40,000	Indus.	15 15 0	Soedel	8/75
Helmholtz Resona- tor Type Gas Dis- charges from Engines	\$ 5,000	Univ.	12 12 0	Soedel	3/76

PROJECT TITLE	AMOUNT OF FUNDING	SOURCE	LEVEL OF EFFORT (MAN-MOS.)	PRINCIPAL INVESTIGATOR(S)	COMPLETION DATE
PURDUE UNIVERSITY - CONTINUED					
Self Excited Oscillations in Gas Flames	\$ 51,000	ASHRAE & AGA	4 24 0	Leonard, Goldschmidt	12/76
Diesel Engine Noise Study	\$ 48,800	NSF	4 24 0	Crocker	1974
In-Duct Fan Research	\$ 1,700	ARI	1/2 1 0	Crocker	1974
Diesel Engine Noise	\$ 10,000	Univ.	0 12 0	Crocker	12/75
Sound Power Measurement	None		0 6 0	Sullivan	2/77
Compressor Noise Study	None		0 6 0	Tree	7/77
Duct Acoustics	\$ 96,500	NASA	5 36 0	Vaidya	6/76
Wind Barriers for Noise Meas- urement	\$ 9,000	Univ.	0 9 0	Tree	12/74
SAN DIEGO STATE UNIVERSITY DEPARTMENT OF AEROSPACE ENGINEERING SAN DIEGO, CALIFORNIA 92182					
Acoustic Char- acteristics of Interacting Supersonic Jets	None		12 12 0	Conly	6/75
UNIVERSITY OF SOUTHERN CALIFORNIA DEPARTMENT OF AEROSPACE ENGINEERING UNIVERSITY PARK LOS ANGELES, CALIFORNIA 90007					
Acoustic and Fluid Dynamic Modelling of the Jet Noise Abatement Problem	\$199,335	DOT	4.6 89 26.6	Laufer	8/76
Basic Noise Research Project	\$ 57,291	NASA	1.4 9.5 14	Laufer	11/77
Test & Advisory Support for Noise & Sonic Fatigue of High Lift Devices	\$106,358	McDonnell, Douglas	1 5 14.8	Kaplan, Van Driest	12/76
STANFORD UNIVERSITY DEPARTMENT OF MECHANICAL ENGINEERING STANFORD, CALIFORNIA 94305					
Internal Flow Noise	\$ 60,000/yr	NSF, Eng.	4 6 0	Johnston	
SYRACUSE UNIVERSITY DEPARTMENT OF MECHANICAL & AEROSPACE ENGINEERING 135 LINK HALL OF ENGINEERING SYRACUSE, NEW YORK					
Noise Reduction from Supersonic Jets	\$225,000	DOT	4 14 0	Dosanjh	10/76

PROJECT TITLE	AMOUNT OF FUNDING	SOURCE	LEVEL OF EFFORT (MAN-MOS.)	PRINCIPAL INVESTIGATOR(S)	COMPLETION DATE
UNIVERSITY OF TENNESSEE DEPARTMENT OF MECHANICAL & AEROSPACE ENGINEERING KNOXVILLE, TENNESSEE 37916					
Study of Fan Noise due to Fluctuating Lift, Inlet Turbulence, and Distortion	\$ 47,060	NASA- Lewis	3 9 0	Lumsdaine	6/77
Investigation of Inlet Noise Suppression Using High Sub- sonic Mach Num- bers	\$ 51,180	NASA- Langley	3 9 0	Lumsdaine	10/76
Research on Compressor Noise and Efficiency	\$ 13,760	NSF	1 5 0	Lumsdaine	6/77
Engineering Science & Mechanics Department Perkins Hall					
Nuclear Power Plant Noise Pre- diction Model for Closed Loop Cooling Systems	\$ 10,000	NRC	2.5 2.5 0	Carley	7/77
TENNESSEE TECHNOLOGICAL UNIVERSITY MECHANICAL ENGINEERING DEPARTMENT BOX 5014 COOKEVILLE, TENNESSEE					
Noise Control Minicourse	\$ 5,000	TDPH	2 0.5 0	Swim	10/75
Noise Control Workshop	\$ 5,000	TDPH	1.5 1 0	Swim	7/76
Design & Con- struction of Noise Control Facility	\$120,000	Univ. Funds, St. of TN, NSF, Indus.	10 6 2	Swim	6/77
F.C. Fan Noise	\$ 15,000	Univ.	6 12 1	Swim & Raj	6/77
Combustion Instability	\$ 62,500	NASA	8 14 8	Purdy	12/75
Combustion Instability	\$ 25,000	NASA	2.5 5 6	Purdy & Ventrice	12/76
UNIVERSITY OF TEXAS AT AUSTIN APPLIED RESEARCH LABORATORIES AUSTIN, TEXAS 78712					
High-Intensity Sound	\$179,000	AFOSS, DOT, NAVSEAS, NASA, ONR	20 52 14	Blackstock	varies
Department of Electrical Engineering					
Acoustic Energy Density Mea- surement	\$ 7,200	Univ.	1 9 0	Hixson	12/76
Vehicle Sound Detection	\$ 42,000	DOT	3 24 0	Hixson	6/75

PROJECT TITLE	AMOUNT OF FUNDING	SOURCE	LEVEL OF EFFORT (MAN-MOS.)	PRINCIPAL INVESTIGATOR(S)	COMPLETION DATE
U.S. NAVAL ACADEMY DEPARTMENT OF MECHANICAL ENGINEERING ANNAPOLIS, MARYLAND 21402					
Coanda Surface Characteristics	\$ 10,200	Navy	4 0 0	Lopardo	10/73
Coanda Flow Resistors	\$ 8,000	Navy	3 0 0	Lopardo	4/76
Coanda/Refraction Noise Suppression Device	\$ 8,000	Navy	3 0 0	Elder	5/76
Coanda Design Parameters	\$ 9,000	Navy	3 0 0	Butler	10/76
Flow Noise Analysis for High Pressure Air & Water Piping Systems	\$ 7,100	Navy	3 0 0	Uldrick	10/76
VILLANOVA UNIVERSITY INSTITUTE FOR TRANSPORTATION STUDIES VILLANOVA, PENNSYLVANIA 19085					
Noise Pollution Study for Traffic Route 29	\$ 4,000	PA DOT	1 4 0	Schuster	7/76
Input to the EIS for the Phoenixville Spur	\$ 45,000	PA DOT	3 8 6	Schuster	7/75
Noise Pollution Study for the Allentown Spur	\$ 4,000	PA DOT	1 4 0	Schuster	10/75
VIRGINIA POLYTECHNIC INSTITUTE & STATE UNIVERSITY MECHANICAL ENGINEERING BLACKSBURG, VIRGINIA 24061					
Prediction of Noise Levels in Manufacturing Areas	\$ 78,000	NSF	11 32 0	Hurst & Mitchell	10/76
Noise Control in the Poultry Processing Industry	\$ 13,090	Indus.	1 9 0	Hurst & Mashburn	9/76
Advanced Acoustic Suppression Concepts	\$ 56,865	NASA	12 12 0	Nayfeh	12/76
Acoustic Properties of a Stretched Membrane with a Sound Absorbing Backing	None		1 6 0	Hurst	7/73
Sound Pressure Level Prediction in Large Rooms	None		1 6 0	Hurst	5/73

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VIRGINIA POLYTECHNIC INSTITUTE & STATE UNIVERSITY - CONTINUED					
A Quantita- tive Depth Perceptor as a Blind Mobility Aid	None	Indus.	1 6 0	Rivers	12/75
UNIVERSITY OF WEST VIRGINIA MECHANICAL ENGINEERING AND MECHANICS DEPARTMENT MORGANTOWN, WEST VIRGINIA 26506					
Design for the Abatement of Air-Abrasive Blast Nozzle Noise	\$ 10,350	Indus.	2 12 0	Sneckenberger	8/74
Sound Transmis- sion through Heated Panels	Unaccounted	Univ.	0.3 4 0	Sneckenberger	5/75
Impact Assess- ment of Phase I PRT Operation on Community Noise	\$ 18,090	US-EPA/ONAC	4 9 0	Sneckenberger	12/76